HYDRO POWER ENGINEERING

JAMES J. DOLAND

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A Textbook for Civil Engineers

By

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PREFACE

This textbook, designed to meet the requirements of a one-semester or quarter course, was written to supply a practical rather than a theoretical approach to the design and preparation of plans for hydroelectric power installations. It has grown out of the author's more than twenty-five years of experience in the design of these plans and teaching them to civil engineering students. The student is made acquainted with the necessary fundamental theory and is then led through the practical routine which he must follow in selecting the proper type and diameter of runner. The discussion then proceeds to the design of water passages, the proper number of units, and finally to the structure itself and the appurtenances essential to its efficient operation. From these designs a preliminary estimate of cost may be derived, as well as the information required by manufacturers of turbines, generators, and accessories.

During the past quarter of a century a great many changes have occurred in the techniques by which a civil engineer proceeds from his fundamental studies of the hydraulic and hydrological potentialities of a site or sites to his preliminary designs. Also within this span of time, the size and capacity of hydraulic turbines have been materially increased, a better understanding of cavitation has been worked out, and improvement in runner types has occurred, particularly within the propeller range. Moreover, the very marked increase in federal activity in hydro power development has had a stimulating impact on the development of new techniques which have simplified traditional approaches to the problem. The growing practice of interconnecting hydro plants with steam electric plants is but one example of the problems with which a civil engineer must now be fully conversant.

Much of the material in this text has been developed from extensive analyses of information received from both public and private sources. The data from public sources were chiefly the practices followed by the Corps of Engineers, the Bureau of Reclamation, and the Tennessee Valley Authority, in both constructed and proposed projects. These agencies, as well as the S. Morgan Smith Company, the Allis-Chalmers Manufacturing Company, the Newport News Ship and Drydock Company, and the Woodward Governor Company have been most cooperative in supplying material which represents the latest practices. In the development of the text, several original studies were made. They

include the equations for runner diameter in terms of specific speed, the outside diameters of generators, the cost of installation of steam plants in dollars per kilowatt, recommended values of plant sigma, and the allocation of powerhouse space. Some of these studies present new or modified techniques for the solution of problems. All are based on actual experience.

The whole field of hydro power engineering will undoubtedly continue to advance rather rapidly. The book recognizes this healthy state of affairs and has purposely been kept small to allow time for direct assignment to the source material which is cited throughout its chapters and to new material as it appears. It is essential that the instructor keep abreast of, and refer his students to, papers dealing with advances and improvements. In most engineering colleges, civil engineering students receive in other courses their training in the design of dams and in electrical and mechanical design. For this reason, these topics have been omitted here. It is assumed that the instructor will correlate his hydro power course with other engineering courses.

The author is deeply indebted to his colleague, Professor Ven Te Chow, for assistance in gathering and drafting the illustrations, for collaborating in the analysis of data, and for his unflagging help in organizing the chapters into their final form.

Numerous individuals and organizations have kindly furnished advice, technical information, photographs, and specific data. In particular, the author wishes to thank Albert S. Fry, Tennessee Valley Authority; John W. Dixon, Bureau of Reclamation; General C. H. Chorpening and Gail Hathaway, Corps of Engineers, Department of Defense; Frank L. Weaver, Federal Power Commission; E. E. Lashway, Allis-Chalmers Manufacturing Company; and Howard A. Mayo, Jr., S. Morgan Smith Company.

JAMES J. DOLAND

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CHAPTER 1

WATER POWER DEVELOPMENT

1-1. Introduction. Since 1920, the installed hydro capacity in the United States has increased from 3.7 million kilowatts to about 19 million kilowatts, or more than 500 per cent. On January 1, 1952, the hydro capacity was about 60 per cent greater than it was on January 1, 1942. This decade showed an increase from 11.8 million kilowatts to 18.9 million kilowatts. The sharp increase was due almost entirely to the activity of the federal government in developing large hydro power sites. Federal plants under construction and authorized for future construction are expected to add 16,988,000 kilowatts.* The federal program is scheduled to almost double the total hydro capacity as of January 1, 1952, not including any private development that might occur. The estimated *undeveloped* water power capacity in the United States is 85 million kilowatts, based on mean flow of streams of which 55 million kilowatts are available 50 per cent of the time. The estimated total available water power in the world is 500 million kilowatts, of which about 60 million, or approximately 12 per cent, has been developed. These statistics are indicative of recent and probable future activity in the water power field. A large part of the engineering work connected with hydro developments is done by civil engineers. The purpose of this text is to cover the civil engineering aspects of the subject which are related primarily to the power plant.

The largest water power developments in the United States are federally owned. Table 1-1 gives a list of the installed capacity at the five largest installations.

TABLE 1-1

LARGEST FEDERAL DEVELOPMENTS (UP TO 1950)

Name	Capacity, in kw
Grand Coulee	1,944,000
Hoover	1,034,000
Bonneville	518,400
Wilson	436,000
Shasta	379,000

* *A Water Policy for the American People,* Report of the President's Water Resources Policy Commission, 1950, Vol. I (Washington, D. C.: Government Printing Office, 1950), p. 222.

The chief federal agencies concerned in the construction of water power plants are the Tennessee Valley Authority, the Corps of Engineers of the U. S. Army, and the Bureau of Reclamation. (See Table 1–2.) The Federal Power Commission is not a construction

TABLE 1–2

ESTIMATED TOTAL HYDRO CAPACITY INSTALLED OR UNDER CONSTRUCTION BY FEDERAL AGENCIES (1952)

	Capacity, in kw
Bureau of Reclamation	6,548,000
Corps of Engineers*	3,790,000
Tennessee Valley Authority	2,656,000
Total	12,994,000

* The Corps of Engineers developed a part of the Wilson Dam installation which is included here in the TVA figure.

agency but exercises supervision over all water power developments, both private and public.

1–2. The Power Market. Figure 1–1 shows the capacity of generating plants in the United States up to the end of 1951.* The total capacity was 75,774,725 kw, of which 18,868,107 kw was in hydro plants. In Fig. 1–2 we see that the production of electric energy in 1951 was slightly less than 371 billion kilowatt-hours, of which approximately 100 billion kilowatt-hours was produced by hydro plants. The five leading states in order of their 1951 total production of electric energy were New York, California, Pennsylvania, Ohio, and Illinois. The five leading hydro states were Washington, California, New York, Tennessee, and Alabama. The 1950 Report of the President's Water Resources Policy Commission says the following about future hydroelectric power development:

"The Federal Power Commission estimates that by 1970, the nation will require a total installed central station power capacity of 160 million kilowatts to supply total energy requirements of 725 billion kilowatt-hours a year. That means an increase of about 93 million kilowatts of capacity and 400 billion kilowatt-hours per year of electric energy over the next 20 years. This tremendous increase in power requirements will be met in part by water power and in part by steam

* Data taken from *Production of Electric Energy and Capacity of Generating Plants*, Report of the Federal Power Commission, 1951 (Washington, D. C.: Government Printing Office, 1951), p. vii.

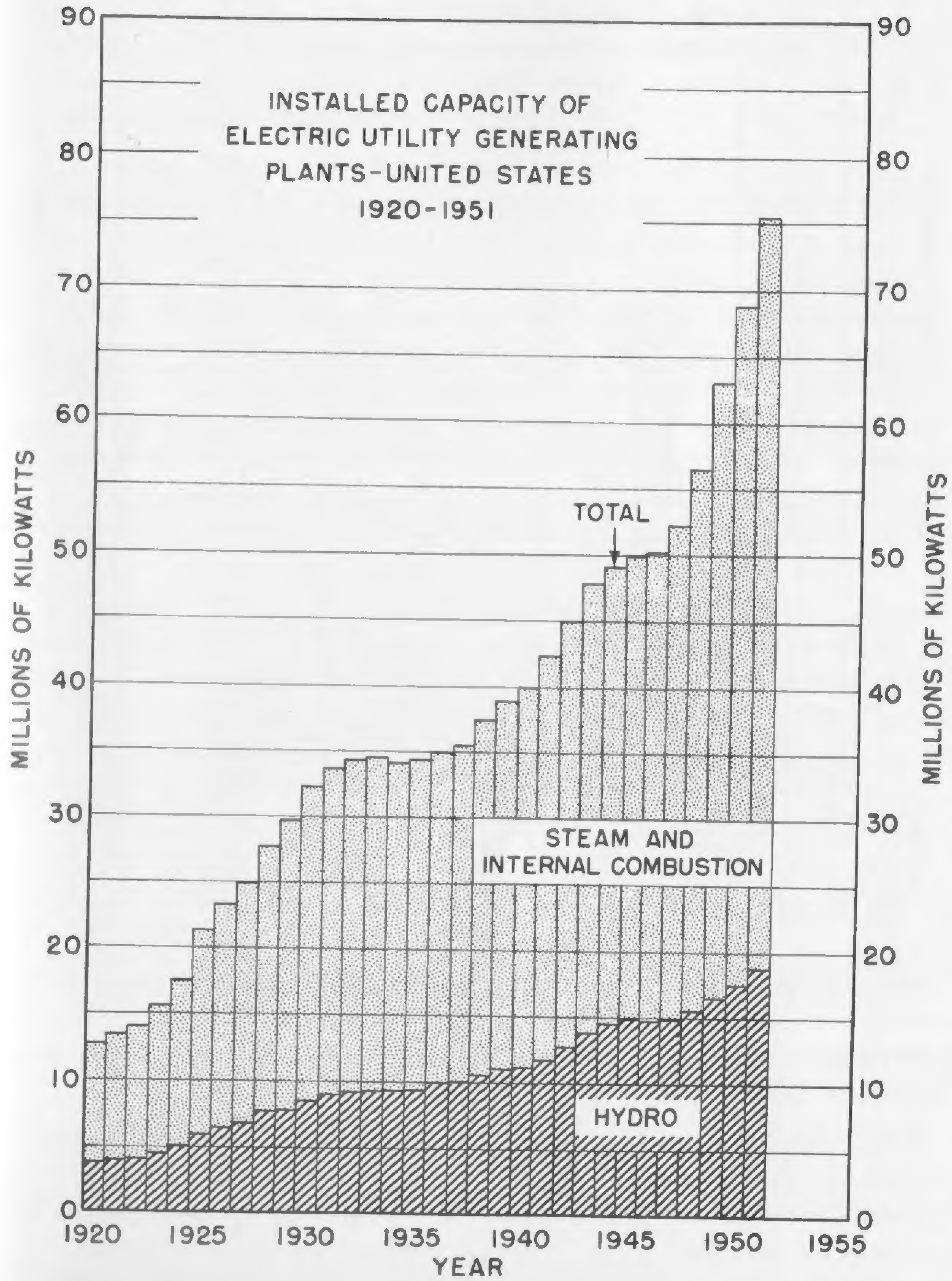


FIG. 1–1. Installed capacity of electric utility generating plants—United States, 1920–51. (Federal Power Commission)

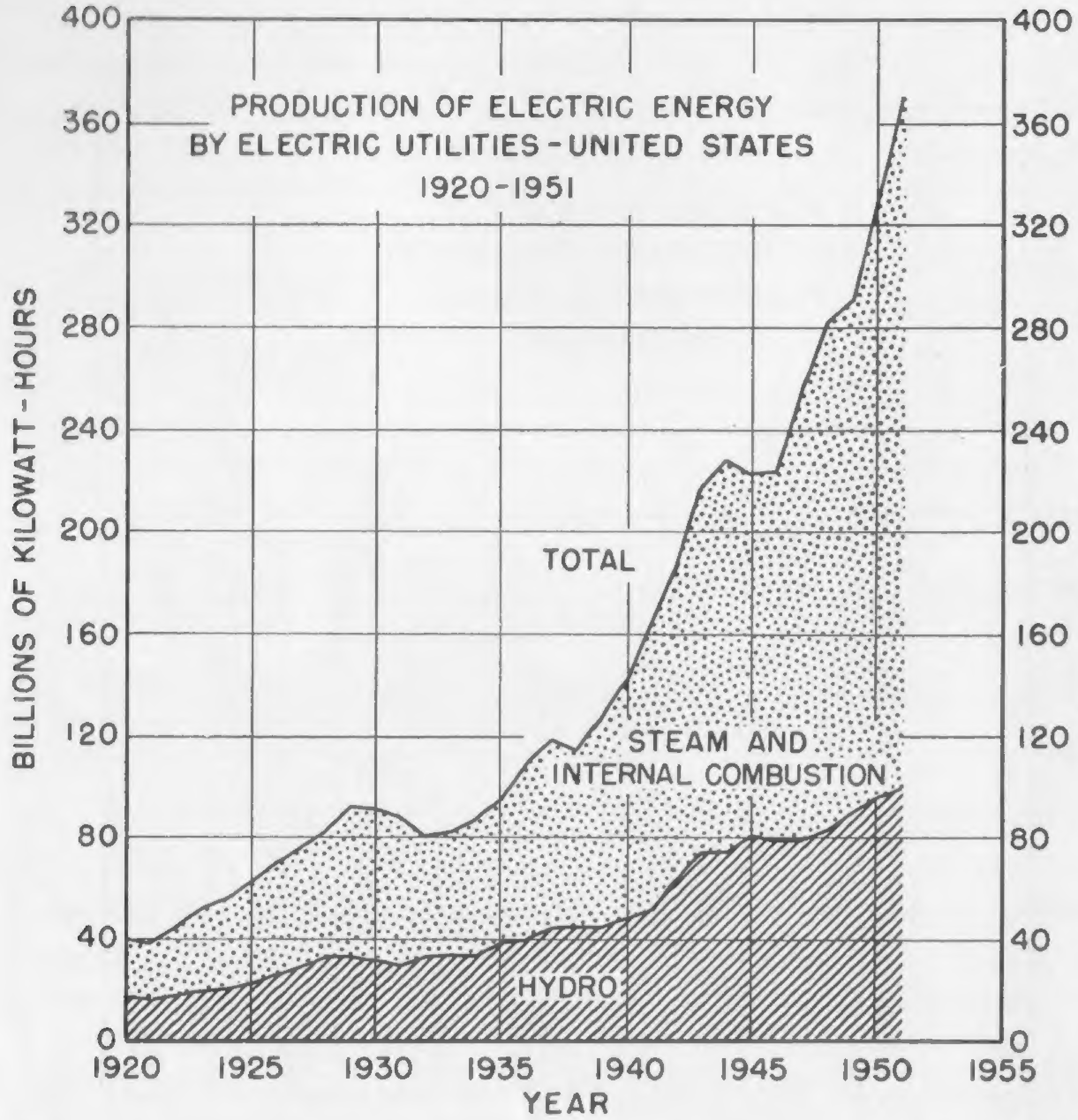


FIG. 1–2. Production of electric energy by electric utilities—United States, 1920–51. (Federal Power Commission)

(fuel). If present practices, based on the most economical use of water power, are followed, this will mean the development of about 25 million kilowatts additional hydro electric power and 68 million kilowatts additional steam (fuel) plant capacity." *

1–3. Annual Plant Factor. The annual plant factor, or capacity factor, is defined as the amount of total production in kilowatt-hours divided by the total possible production in kilowatt-hours, expressed in per cent. The annual plant factor in 1951 for all United States plants was 58.5 per cent; for fuel plants, 57.2 per cent; and for hydro plants, 62.3 per cent. In 1936 the corresponding annual plant factors were: all

* *A Water Policy for the American People,* Report of the President's Water Resources Policy Commission, 1950, Vol. I (Washington, D. C.: Government Printing Office, 1950), p. 239.

U. S. plants, 35.8 per cent; fuel plants, 31.9 per cent; hydro plants, 45.8 per cent. These figures are indicative of the improved use of generating plant capacity.

Hourly, daily, and seasonal load variations, together with the need for periodic overhauling of equipment, make the attainment of a 100 per cent annual plant factor impossible.

1–4. Water Power and River Basin Development. The trend in the United States today is toward the integrated development of the total water resources of river basins. The large structures now being built usually serve multiple purposes, such as irrigation, flood control, flow regulation, navigation, recreation, and water power. Water power, where economically feasible, becomes one of the important components in a conservation program. It not only provides a source of cash income, but it may also contribute to the social well-being of the region. The Tennessee River Valley is an example of coordinated water resource development. Here water power plays a leading role, although the need for associated steam plants suggests that fuel plant capacity in 1954 could exceed the hydro capacity by 500,000 kilowatts. Figure 1–3 shows a pictorial diagram of the Tennessee Valley Authority water control system, and Fig. 1–4 shows a map and profiles of the streams in the region. Water power turbines are installed in connection with most of the dams. All of the dams serve flood control, flow regulation, and recreational purposes. In addition, the main stream dams provide navigation facilities.

The Bureau of Reclamation has provided many water power installations in connection with irrigation structures. The Corps of Engineers builds water power plants in structures constructed primarily for flood control and navigation improvements.

1–5. Hydro Power Development by Private Enterprise. The tremendous increase in the construction of hydroelectric developments has spurred private power companies to become more interested than previously in the development of sites of large potential capacity. The motivation for this interest stems in part from the fear of nationalization of the power industry by the federal government. Examples of activities by private enterprise are as follows:

1. The Cabinet Gorge plant in Idaho. Completed, with a capacity of 200,000 kw.
2. The Niagara Falls development in New York. In 1952 five major utility companies were seeking rights to develop the first stage, or a capacity of 335,000 kw, within a period of three years after beginning of construction. Development was to be

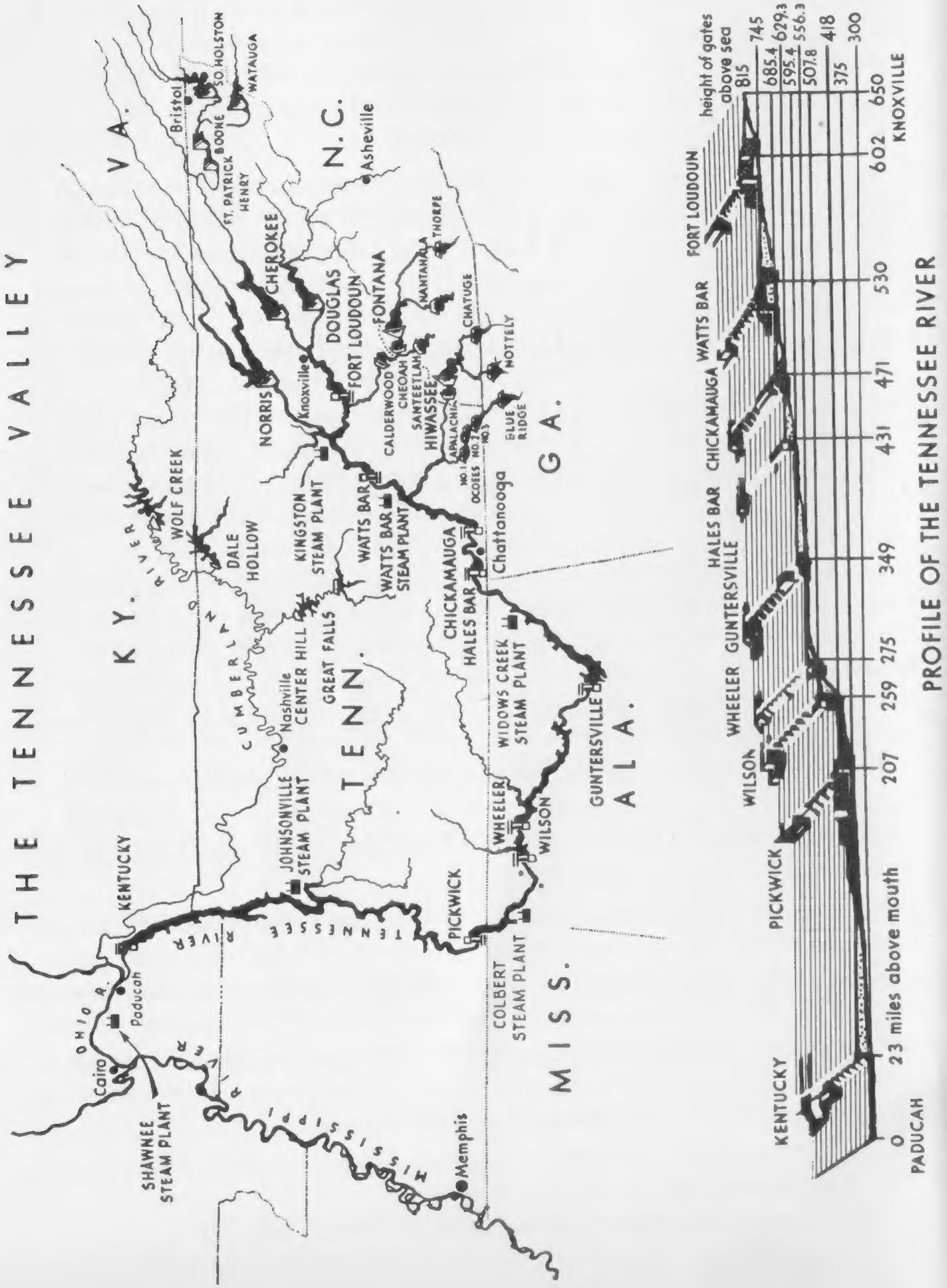


Fig. 1-3. Map of Tennessee River and tributaries. (Tennessee Valley Authority)

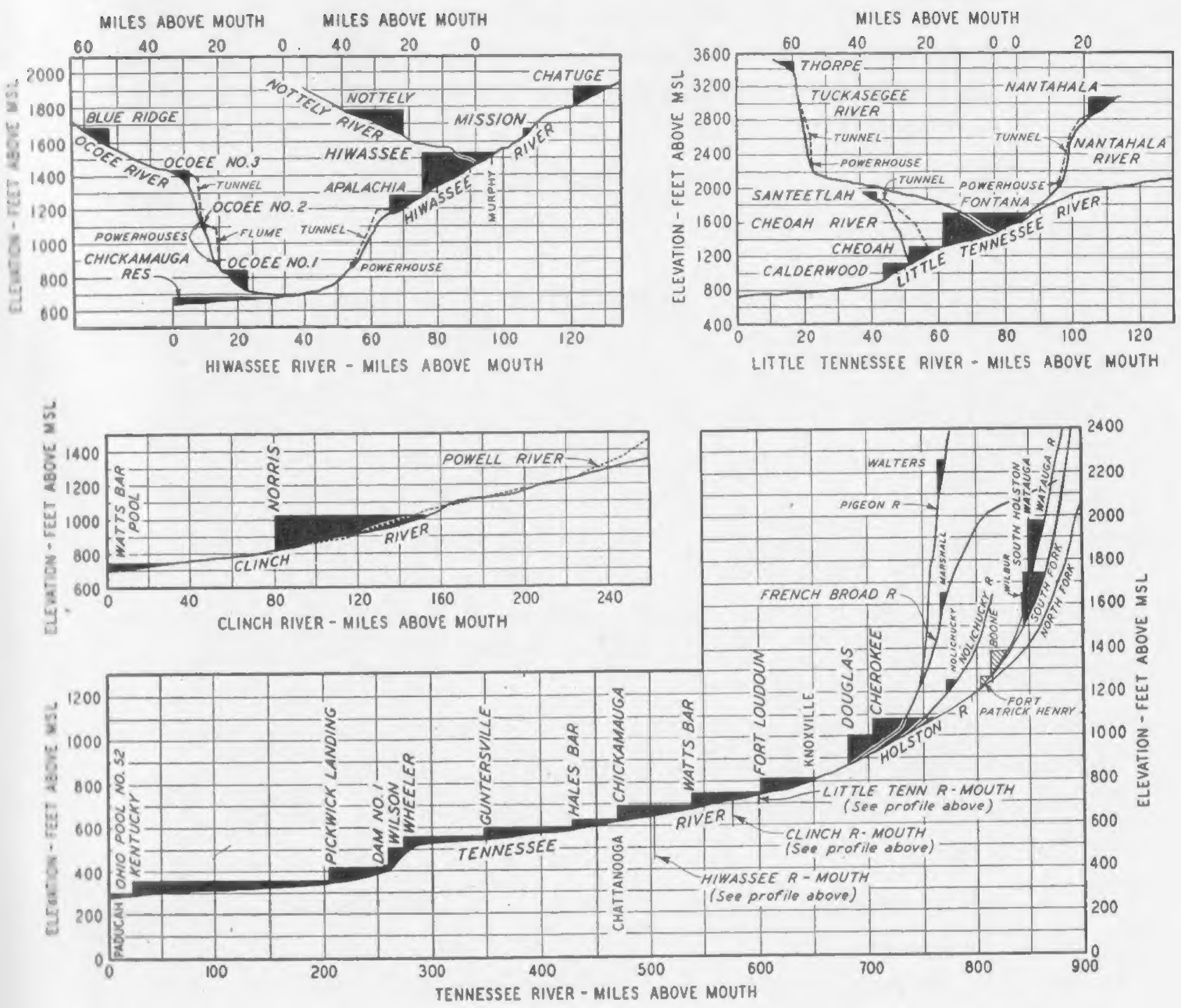


Fig. 1-4. Profiles of Tennessee River and tributaries. (Tennessee Valley Authority)

done without federal subsidy and was expected to produce \$23,000,000 per year in new taxes. Three bills were introduced in the United States 82nd Congress in connection with this project.

a) The Capehart-Miller Bill (S. 2021, HR 3146) would permit the development by private enterprise without cost to federal or state taxpayers.
b) The Lehman-Roosevelt Bill (S. 517, HR 1642) which would empower the federal government to construct the project with public funds and eventually transfer the project to the state of New York.
c) The Ives-Cole Bill (S. 1963, HR 5099) proposes construction and operation by the state of New York with funds derived from tax-free revenue bonds.

3. The Roanoke Rapids project in North Carolina. This project is expected to cost $27,000,000. In 1952 a private company and the Department of the Interior were both seeking rights to build this project.

The latter two projects raise political issues in which all engineers should be interested.

1-6. The Federal Power Commission. The Federal Power Commission plays a very important role in the licensing and regulation of potential and developed hydro power installations in the United States. When first created by Congress in 1920 to administer the provisions of the Federal Water Power Act of 1920, it was composed of the Secretaries of War, Interior, and Agriculture. In 1930 Congress approved an act which organized the Commission as an independent agency administered by five full-time commissioners, appointed by the President and confirmed by the Senate. The term of office of the Commissioners is five years. The Public Utility Act of 1935 extended the jurisdiction of the Commission over electric utilities. The activities and personnel of the Commission have been greatly expanded since 1920 through various congressional acts and executive orders.

The staff of the Commission is now composed of the following offices, bureaus, and divisions:

1. Office of the Secretary
    - Divisions
        - Personnel and Administrative Services
        - Budget and Finance
        - Publications
2. Office of the Chief Engineer
3. Bureau of Power
    - Divisions
        - Licensed Projects
        - River Basins
        - Electric Resources and Requirements
        - Projects Cost
4. Bureau of Accounts, Finance, and Rates
    - Divisions
        - Accounts
        - Original Cost
        - Rates
        - Finance and Statistics
        - Gas Certificates

5. Bureau of Law
    - Divisions
        - Electric Power
        - Natural Gas
        - Interpretation and Research
        - Hydroelectric and Licensed Projects
        - Examiners

The Office of the Secretary handles administrative and housekeeping functions. The Office of the Chief Engineer advises the Commission on engineering matters. The Bureau of Power advises on electric power problems, licensing, and multiple-purpose development. The Bureau of Accounts, Finance, and Rates advises on rate regulation, accounting and financial practices, certificates of convenience and necessity, and other related matters. The Bureau of Law advises on all legal questions; it represents the Commission at hearings and in court cases.

The basic objectives of the Water Power Act of 1920 were to safeguard the ownership and exploitation of water resources and to promote their full development. The Federal Power Commission was charged with the responsibility not only for granting licenses for water power development, but in addition to collect information and data relating to the water resources of the nation. The Bureau of Power is the agency which executes these functions principally through its Division of Licensed Projects and the Division of River Basins. Licenses, when approved, may be granted to individuals, private corporations, or public agencies. They are issued for a period not to exceed 50 years, at the end of which the development is subject to recapture by the United States.

The Commission as a whole administers the various power acts and portions of other statutes. It functions principally as a policy-making board. It grants licenses, issues rules for guidance of the electric power and natural gas industries, and develops the policies to be followed by its staff. The results of its actions are issued in the form of orders and opinions.

CHAPTER 2

THE FUNDAMENTALS OF WATER POWER STUDY

In order to familiarize himself with the subject of water power engineering, the student must first become acquainted with certain basic principles, mathematical relationships, and definitions of terms. This chapter is intended to provide an introduction to these relationships and terminology.

2–1. Water as a Source of Power. Water in nature is considered a source of power when it is able to perform useful work—particularly turn water wheels and generate electricity—at a rate such that the delevopment of power can be accomplished in a most efficient and economical way.

Water has the capacity to perform work if it possesses energy. Water used for power development purposes, such as in streams, rivers, and lakes, derives its energy from the radiant energy of the sun through the operation of the hydrologic cycle. In the hydrologic cycle, water from ground and water surfaces is evaporated by solar radiation into the atmosphere. This vapor is cooled and condensed by various meteorological processes and falls to the earth as precipitation. Some of this precipitation falls directly on the sea and some falls on land surfaces. Part of the precipitation falling on highlands will flow as runoff to streams or lakes, where water may be collected in sufficient quantity and, if it possesses enough potential energy, may be utilized for power generation.

As water flows from highlands to a lower elevation, its potential energy is reduced by evaporation, drop in elevation, and other causes. A part of this reduced amount of potential energy may be converted to mechanical energy and used to generate power. The remaining part of the potential energy is dissipated in the form of heat through friction and turbulence, which occur in flowing streams and in the water passing through structures and power plant machinery. The energy dissipated in such ways, called losses, cannot be recovered and utilized for power generation at the particular site. Therefore, the efficient development of energy in water depends on the technique of minimizing such losses.

The amount of water power developed from any stream, river, or lake is measured primarily by (1) the rate of discharge of water and (2) the head that is available. Both the rate of discharge and the head are variable quantities which may fluctuate at any moment and at any

point in a stream. These variations vitally affect the power that can be developed by a plant designed and installed for power generation purposes. Consequently, an accurate determination of these quantities is essential to the successful design of a water power project.

In making preliminary estimates of the potential power production of streams, the total available electric energy may be determined by the following commonly used formula:

$$E = 1.025VHe \tag{2–1}$$

where E is the energy in kilowatt-hours; V is the total volume of water in acre-feet; H is the static head in feet; and e is the over-all efficiency, which may be taken as 80 per cent for storage plants and 75 per cent for run-of-river plants (see Art. 2–5).*

2–2. The Water Power Equation. If a steady discharge of Q cubic feet per second (cfs) is available with a net head of H feet, the power that can be developed from this quantity of water passing through a power generating installation, expressed in horsepower and kilowatts, respectively, will be

$$P(\text{hp}) = \frac{QHwe}{550} \tag{2–2a}$$

$$P(\text{kw}) = \frac{QHwe}{737} \tag{2–2b}$$

where w is the unit weight of water in pounds per cubic feet, e is the efficiency of the power generating installation, 550 is the number of foot-pounds per second in one horsepower, and 737 is the number of foot-pounds per second in one kilowatt. Since $w = 62.5$ lb per cu ft, then Eqs. (2–2a) and (2–2b) become:

$$P(\text{hp}) = \frac{QHe}{8.8} \tag{2–3a}$$

$$P(\text{kw}) = \frac{QHe}{11.8} \tag{2–3b}$$

With an assumed average efficiency of a power plant at 88 per cent, the power Eq. (2–3a) becomes:

$$P(\text{hp}) = \frac{QH}{10} \tag{2–4}$$

This is a convenient formula which may be used for quickly estimating the horsepower that can be developed by a water power or hydroelectric plant.

* *Bureau of Reclamation Manual*, Vol. IV (Washington, D. C.: Government Printing Office, 1948), par. 4.2.2.

2–3. Quantity of Water. The source of the water supply for hydroelectric plants is, of course, the rainfall. Since rainfall is quite variable in quantity and occurrence, the resulting runoff is by no means constant. The demand on power resources is usually much more uniform than natural stream flow. Hence regulation of the natural supply is desirable when economically feasible from an engineering viewpoint. There are cases in which natural regulation of flow takes place through the presence of large lakes. This situation occurs on the lower St. Lawrence River, where flow is regulated by the Great Lakes. Regulation of flow is also accomplished by artificial storage reservoirs for long-term regulation, and by pondage for the regulation of hourly, daily, or weekly flows. Pondage consists of a relatively small reservoir located at or near the plant site. The subjects of storage and pondage are treated in Chapter 3.

The methods of determining the discharge in streams flowing in open channels are numerous and varied. Two extensively used methods are the current meter method and the weir method. The current meter method is suitable for all rivers under different flowage conditions and at stream sections, whereas the weir method is reliable and satisfactory for use in open channels when the obstruction of the channel and change of water levels, due to the interposition of the weir, are not objectionable. However, if possible, it is always advantageous to use the stream flow records available from the U. S. Geological Survey or from state or local agencies. A daily record of flow covering a period of ten years or more will provide reasonably good information if the data are reliable. The data should be properly adjusted to include any possible artificial or natural change in the stream regimen that has occurred within the period, such as the construction of reservoir or other channel regulation devices upstream, alteration in water courses, and unusual climatological phenomena.

The analyses required regarding stream discharge for the investigation of a power project should generally cover the following items:

1. The flow duration curve at the plant site as a basis for the determination of plant capacity and power supply at all times. The flow duration curve is a graph showing the flow in cubic feet per second plotted against the per cent of time for which a given flow is equaled or exceeded during a specific period of record. (See Fig. 3–1.)
2. Minimum flow as a basis for the determination of firm power and the design of plant auxiliaries. (See Art. 3–5.)
3. Flood flows as a basis for the proper design of spillway and other flood regulation measures to provide for the safety of the plant.
4. The mass curve of runoff at the plant site as a basis for the study of storage effect and flow regulation. The mass curve is a graph showing the accumulated volume of water which has passed a particular station plotted against time. (See Fig. 3–3.)

2–4. Head of Water. Various modifying terms are used in connection with the term *head*. The student is apt to become confused when he attempts to determine the exact meaning of the terms as he encounters them in the literature. The following definitions are most commonly employed and will be used for purposes of this text, even though some of them are not universal.

The *gross head* is the difference in elevation between the water surface in the forebay or reservoir above the plant and the water surface in the tailrace below the plant when the turbines are not operating. The gross head may vary widely due to variations in stream flow conditions and reservoir levels. In some instances, the gross head may approach zero during periods of flood flow. When reference is made to gross head, it is advisable to specify the conditions of flow by giving maximum, minimum, and average, or normal, values of the gross head. The statement of conditions of flow is particularly applicable to a low-head plant that is subject to flood flow conditions, as well as to all types of plants operated from a multi-purpose reservoir, upon which functions of flood control, water supply, and recreational limitations are imposed as operational restrictions.

The *net*, or *effective, head* for reaction turbines is defined by the American Society of Mechanical Engineers as "the difference between the total energy contained in the water immediately before its entrance into the turbine and its total energy immediately after discharge from the draft tube"; or "the effective head on the turbine shall be taken as the difference between the elevation corresponding to the pressure head in the penstock at the entrance to the turbine casing and the elevation of tailwater, the above difference being corrected by adding the velocity head in the penstock at the section of measurement and subtracting the residual velocity head at the section of measurement in the tailrace." * This means that all hydraulic losses other than those which occur within the turbine and its parts are not chargeable to the performance of the turbine with respect to its efficiency.

For impulse turbines, the effective head is roughly determined by subtracting from the elevation of the pressure head, plus velocity head

* American Society of Mechanical Engineers, *Test Code for Hydraulic Prime Movers* (New York: The Society, 1949), p. 17; *ibid.*, p. 18.

at a point immediately upstream from the nozzle, the elevation of the lowest point of the pitch circle of the runner buckets.

Design head is the effective head for which the turbine is designed by the manufacturer for best speed and efficiency. Generally the design head is selected as that head above and below which the average annual generation of power is approximately equal. This selection will tend to provide the most efficient use of water by having the point of best efficiency at the weighted average head. The maximum head under which a Francis turbine operates should not exceed 125 per cent of the design head, and the minimum head should not be less than 65 per cent of the design head. Corresponding values for fixed-blade propeller types are 110 per cent and 90 per cent; for movable-blade propeller types, 150 per cent and 50 per cent.

Rated head is the effective head at which the full gate output of the turbine will produce the rated capacity of the generator in kilowatts. It is also the effective head at which the horsepower of the turbine is guaranteed by the manufacturer.

Critical head is the effective head which determines the elevation of the turbine runner with respect to the elevation of tailwater, known as the setting of the turbine. The position of the runner with respect to the tailwater may control the phenomenon of cavitation. This phenomenon is discussed in Chapter 4. The use of the term critical head is rather restricted at the present time but may become more prevalent in the future. In support of the use of this term a Tennessee Valley Authority technical monograph states: "There appears to be no reason why turbine dimensions as well as turbine setting should not be referred to this critical head. Confusion of using one head for the determination of turbine sizes and another head for the determination of turbine setting will thereby be avoided." *

2–5. The Efficiency of a Hydroelectric Installation. Efficiency is the ratio of the power delivered by a machine or other apparatus to the power delivered to it, usually expressed as a percentage. This is always less than unity, or 100 per cent, because losses in the generation and transformation of energy of a hydroelectric development are inevitable. (See Figs. 4–2 and 4–3.)

Theoretically speaking, the over-all efficiency of a plant is the product of the instantaneous efficiencies of all its elements. These elements include the turbine and its water passages (scroll case and draft tube), generator, transformers, transmission lines, and other apparatus

* W. L. Voorduin, *Preliminary Selection of Hydraulic Turbines and Powerhouse Dimensions*, TVA Technical Monograph No. 52 (rev. ed.; Knoxville, Tenn.: Tennessee Valley Authority, June, 1945), p. 10.

through which the power is transmitted. However, it seems to be more practical to compute the efficiency as the ratio of the power output of the plant to the water power available at the plant site. It should be noted that the efficiency of a plant or plant element is different under different service conditions. For example, when the over-all plant efficiency is given, it may be the efficiency at which the discharge through the plant is at maximum full-gate point or the efficiency at which the turbine efficiency is the best full-load point. In all cases the condition at which the efficiency is defined should be clearly understood. *Full gate* is defined here as that condition which obtains when the turbine gates are fully open. *Full load point* is the prevailing condition when the turbine gates are set at a position which produces maximum efficiency or minimum hydraulic losses within the turbine.

2–6. Types of Hydraulic Turbines. The hydraulic turbine is the prime mover of a water power development. It is a rotary engine actuated by the reaction or impulse, or both, of a current of water. The revolving part of such an engine is called the runner, which consists of a series of curved vanes, blades, or buckets, mounted on a central shaft.

The hydraulic turbines employed in modern practice may be classified into two general groups: the reaction turbine and the impulse turbine. In the reaction turbine the runner chamber is completely filled with water, which acts by its reactive pressure. In the impulse turbine the head is converted into velocity, producing kinetic energy, and the runner chamber is not completely filled with water. The reaction turbine is of two main types: the Francis and the propeller turbines. The former is subdivided into high-speed, medium-speed, and low-speed Francis turbines; and the latter, into fixed-blade and movable-blade propeller turbines.

In the Francis turbine the runner is constructed so that the water passes through it in a direction that is first radial and then axial. The inlet diameter of the runner is greater than the discharge diameter for high-speed runners (low specific speed) (Art. 2–13, Fig. 2–1A); about equal for specific speeds near 40 (Fig. 2–1B); and smaller than the discharge diameter for low-speed runners (high specific speed) (Fig. 2–1C). The inlet diameter is usually referred to as the nominal diameter of the Francis turbine. (See Art. 2–7.)

The propeller turbine is so constructed that the water passes through the runner in an axial direction. The blades of the runner may be fixed or movable. The best known of the movable type is the Kaplan turbine (Fig. 2–2), the blades of which can be rotated or adjusted to such a position as to attain the best efficiency under various operating conditions. (See Chapter 4.)

Fig. 2-1B. Medium-speed Francis-type runner; $N_s = 35$; $N = 107$ rpm; head = 214 ft; $P = 70,000$ hp. (Allis-Chalmers Manufacturing Co.)

Fig. 2-1A. High-speed Francis-type runner: $N_s = 17$; $N = 720$ rpm; head = 550 ft; $P = 4000$ hp. (Allis-Chalmers Manufacturing Co.)

Fig. 2-1C. Low-speed Francis-type runner: $N_s = 66$; $N = 100$ rpm; head = 92 ft; $P = 35,000$ hp. (Allis-Chalmers Manufacturing Co.)

The impulse turbine (Fig. 2-3) is operated in such a way that the runner is moved by a jet of water issuing from the orifice of a nozzle. It is generally classified as single-jet type or multiple-jet type, according to the number of jets provided for each runner. The best-known type of impulse runner is the Pelton type.

2-7. Designation of Runner Diameters. Figure 2-4 shows the method of designating runner diameters. The nominal diameter is the diameter of the runner of a Francis turbine measured at the center line of the distributor. It is designated by D_1. The minimum diameter, measured as the minimum diameter of the inside of the shroud, is designated by D_2. The discharge diameter of the runner is measured at the bottom of the shroud and is designated by D_3. The ratio D_3/D_1 can be determined approximately by the formula

$$D_3/D_1 = 0.01N_s + 0.54 \tag{2-5}$$

where N_s = specific speed

FIG. 2-2A. Shop assembly of Kaplan-type runner: $N_s = 125$; $N = 125$ rpm; head = 69.2 ft (range 54.8 to 105 ft); $P = 40{,}000$ hp. (S. Morgan Smith Co.)

FIG. 2-2B. Installation view of Kaplan-type runner. (S. Morgan Smith Co.)

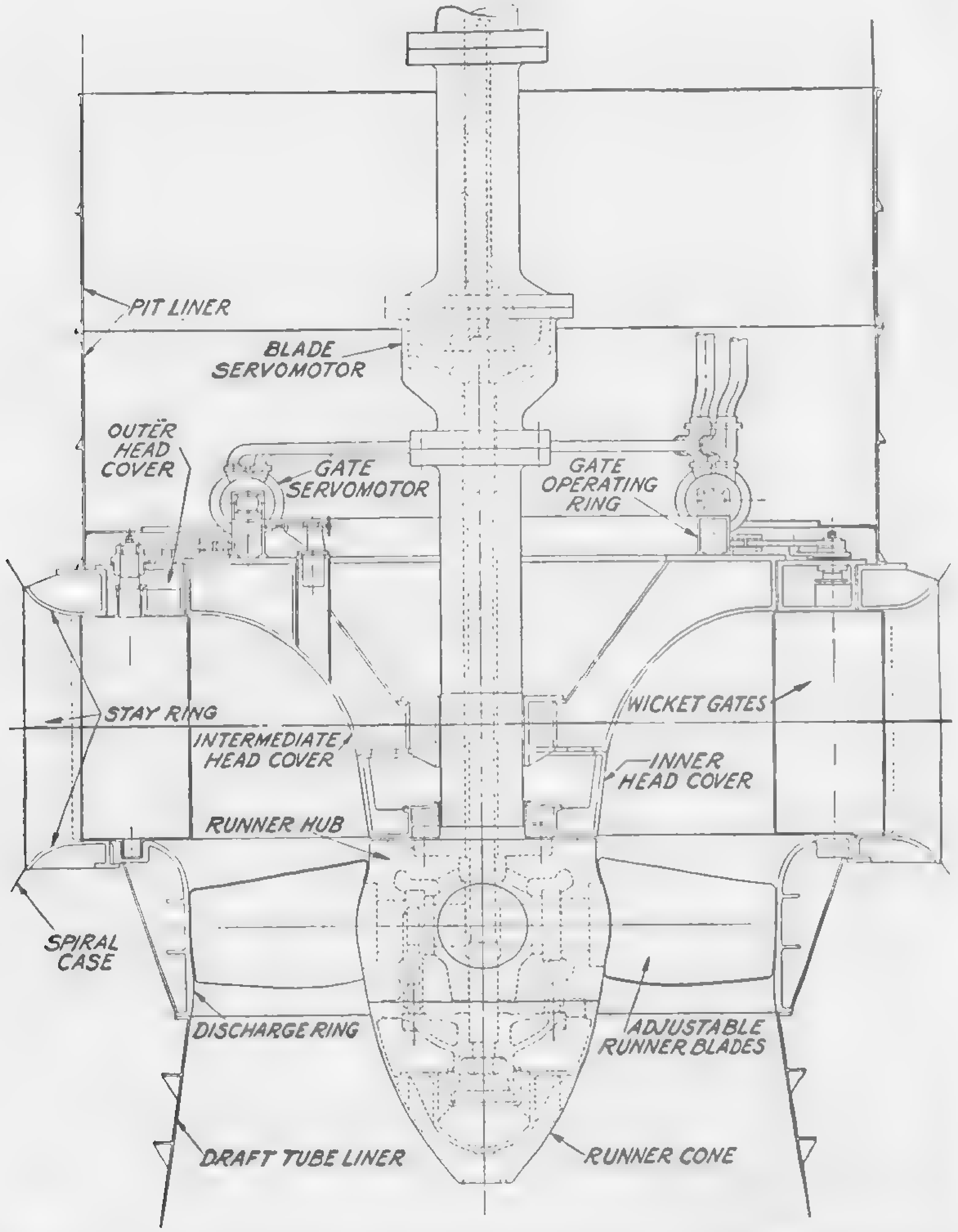


FIG. 2-2C. Typical drawing of Kaplan-type runner. (S. Morgan Smith Co.)

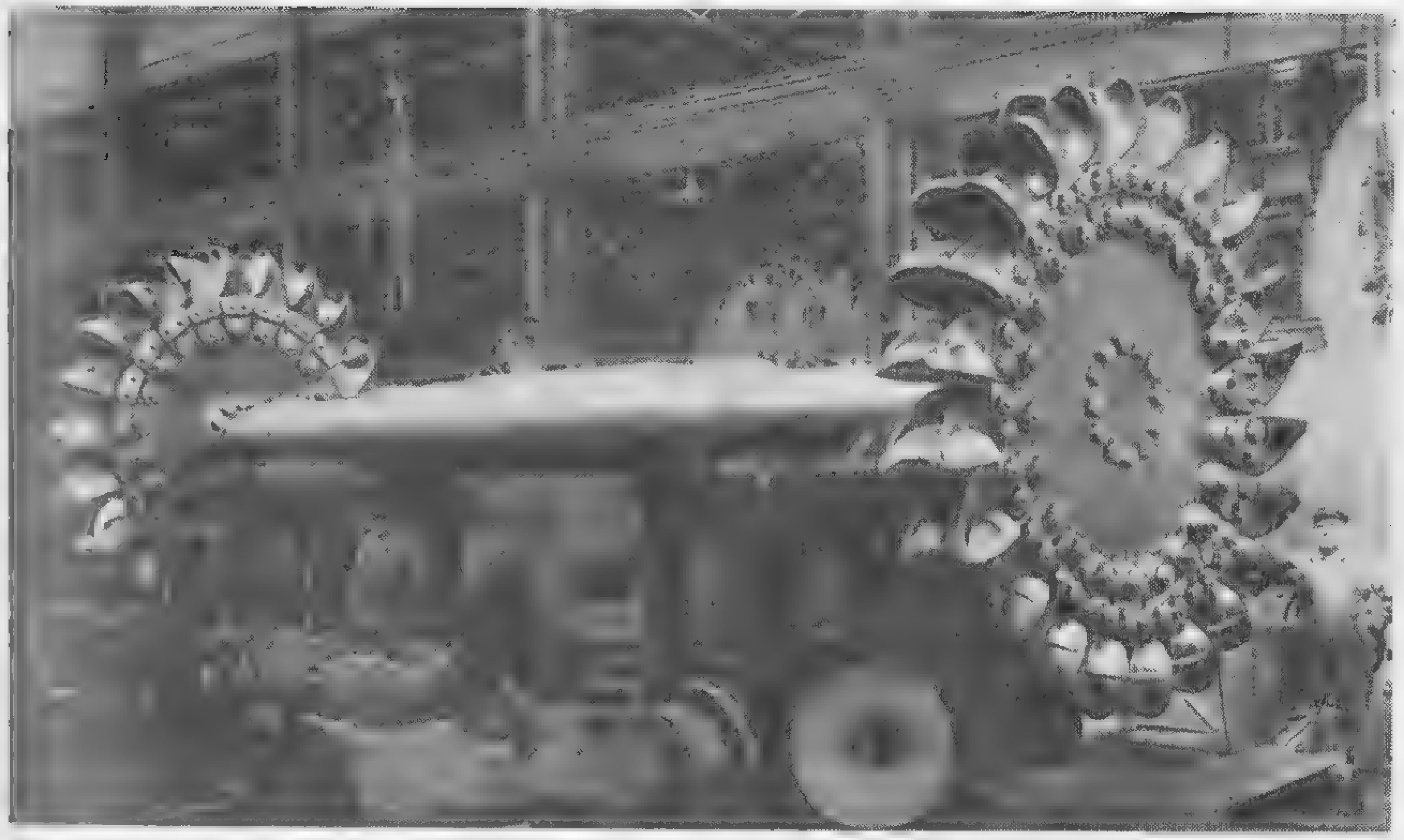


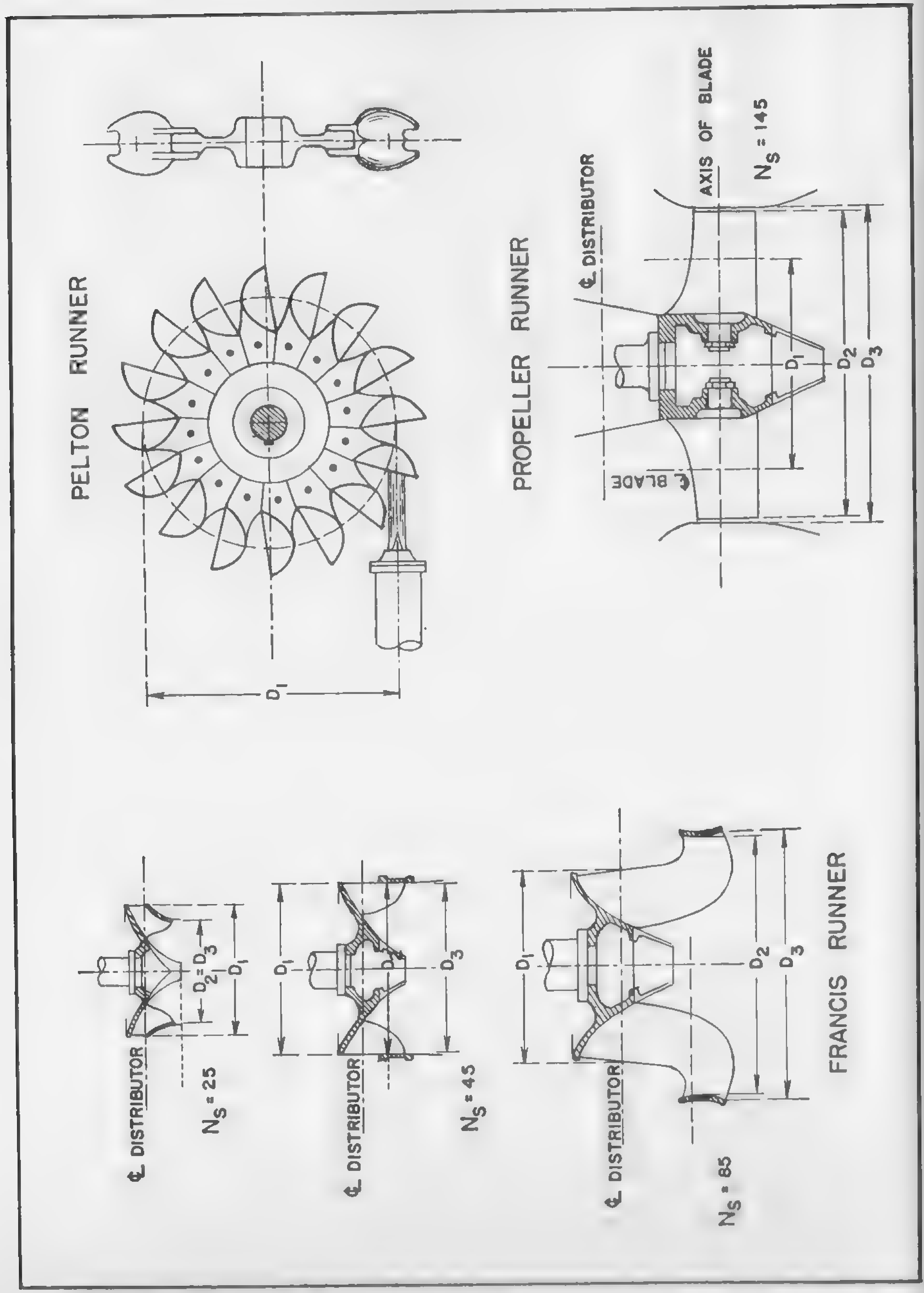
FIG. 2-3. Impulse-type runner: $N_s = 3.2$; $N = 360$ rpm; head $= 2243$ ft; $P = 40{,}000$ hp. (Allis-Chalmers Manufacturing Co.)

The nominal diameter of a propeller is considered as the diameter measured from the outside to outside of the runner blades and is designated by D_2. The outlet diameter is measured at the throat or top of the draft tube and is designated by D_3. (See Fig. 2-4.)

The diameter of an impulse runner is the pitch diameter or diameter of a circle passing through the centers of the buckets. It is designated by D_1.

2-8. Turbine Constants. The turbine constants are the characteristic values of a turbine. If two or more turbines of different sizes are made similar in design and construction so that the corresponding linear dimensions bear a common geometrical ratio, the turbines are said to be homologous. When operating at a given gate and speed, the homologous turbine should demonstrate similar characteristics and possess the same turbine constants as its prototype. These constants, as discussed in the following articles, include the *unit speed, unit discharge, unit power, specific speed,* and *specific diameter.* The condition of operation for which the turbine constants are defined is usually taken at the rated point, i.e., some per cent less than maximum power under normal speed. Occasionally, the turbine constants also apply to the best efficiency point, i.e., the gate opening for best efficiency.

The turbine constants are of great significance and value to the turbine manufacturer and the plant designer. Usually, they are used for the following purposes:



1. To predict within engineering accuracy the characteristics of a full-size runner from the results of the careful test of a homologous model.
2. To compare the characteristics of two or more turbines of the same type, design, and construction, but of different sizes.
3. To serve as an index, referring to the specific speed, for the preliminary design of a plant.

Turbine constants are explained and derived in Arts. 2-9 to 2-14 inclusive.

2-9. Speed Factor. The speed factor, designated by ϕ, is defined as the ratio of the peripheral or linear velocity of the buckets at the nominal diameter to the theoretical spouting velocity of water under the head acting on the turbine. If v is the peripheral velocity in feet per second, and H is the effective head in feet, then

$$\phi = \frac{v}{\sqrt{2gH}} \tag{2-6}$$

Since

$$v = \frac{\pi D_1 N}{12 \times 60} \tag{2-7}$$

where D_1 is the nominal diameter of runner in inches and N is the speed of runner in revolutions per minute, then

$$\phi = \frac{D_1 N}{1840\sqrt{H}} \tag{2-8}$$

2-10. Unit Speed. From Eq. (2-8) the speed of a runner may be expressed as

$$N = 1840\phi\frac{\sqrt{H}}{D_1} \tag{2-9}$$

Since the unit speed, designated by N_1, is defined as the speed of a homologous runner having a diameter of 1 in., operating under a head of 1 ft, it is apparent from Eq. (2-9) that

$$N_1 = 1840\phi \tag{2-10}$$

It is a constant since ϕ has a definite value for a given turbine when run at a given speed. Again, from Eq. (2-9) we obtain the equation for computing the unit speed from the given values of H and D_1 as follows:

$$N_1 = \frac{ND_1}{\sqrt{H}} \tag{2-11}$$

2–11. Unit Discharge. A turbine may be considered as a kind of orifice to which the laws of discharge through orifices under variable head can be applied. The following formula for the discharge through an orifice can therefore be applied as well to a turbine:

$$Q = \frac{C\pi}{4 \times 144} D_1^2 \sqrt{2gH} \tag{2–12}$$

where Q is the discharge in cubic feet per second, C is the coefficient of discharge and constant for a particular turbine, at a particular gate opening, D_1 is the diameter of the orifice or of the runner in inches, and H is the head in feet.

The unit discharge, designated by Q_1, is defined as the discharge of a homologous runner 1 in. in diameter, under a head of 1 ft. Since Q_1 is a combination of all of the constants in Eq. (2–12), we have

$$Q_1 = \frac{C\pi\sqrt{2g}}{4 \times 144}$$

From Eq. (2–12) it is apparent that

$$Q = Q_1 D_1^2 \sqrt{H} \tag{2–13}$$

Solving for Q_1,

$$Q_1 = \frac{Q}{D_1^2 \sqrt{H}} \tag{2–14}$$

2–12. Unit Power. Substituting Eq. (2–13) into Eq. (2–2a),

$$P = Q_1 D_1^2 H^{3/2} \frac{we}{550} \tag{2–15}$$

As the unit power, designated by P_1, is defined as the power developed by a homologous runner 1 in. in diameter under a head of 1 ft, it is apparent from Eq. (2–15) that

$$P_1 = \frac{Q_1 we}{550} \quad \text{or} \quad \frac{Q_1 e}{8.8} \tag{2–16}$$

This equation corresponds to Eq. (2–3a) when $H = 1$ ft and $Q = Q_1$. P_1 is a constant if the efficiency e is assumed to be constant for all homologous turbines operating at a given gate and speed. The assumption of constant efficiency is not entirely valid because of scale effect. Turbines geometrically similar but having different dimensions are not completely homologous because of differences in interior surface roughness and bearings. These differences will cause variations in the proportional losses of head due to hydraulic and mechanical frictions which in turn will result in changes in efficiency between the two turbines.

Moody * developed the following formula to express the relation between the efficiencies of model and prototype turbines:

$$e' = 1 - (1 - e)\left(\frac{D}{D'}\right)^{1/4} \tag{2–16a}$$

in which: e' is the efficiency of the larger or prototype turbine having a diameter D'; and e is the efficiency of the smaller or model turbine having a diameter D_1. For computing the unit power from given values of P, D_1 and H, the following equation is obtained from Eqs. (2–15) and (2–16):

$$P_1 = \frac{P}{D_1^2 H^{3/2}} \tag{2–17}$$

2–13. Specific Speed. From Eqs. (2–2a), (2–9), and (2–12) the following relations may be found:

$$P \propto QH; \qquad N \propto \frac{H^{1/2}}{D_1}; \qquad Q \propto D_1^2 H^{1/2} \tag{2–18}$$

Eliminating Q and D_1 from the above expressions and solving for N, we get

$$N \propto \frac{H^{5/4}}{\sqrt{P}} \tag{2–19}$$

or

$$N = (\text{a constant}) \times \frac{H^{5/4}}{\sqrt{P}} \tag{2–20}$$

The specific speed is defined as the speed of a homologous runner when it has been so reduced in size that it develops 1 hp under 1-ft head. It is apparent from Eq. (2–20) that the constant in the equation is the specific speed, or

$$N_s = \text{a constant} = \frac{N\sqrt{P}}{H^{5/4}} \tag{2–21}$$

The specific speed is a value of considerable significance in the preliminary design of a power plant. Its notable applications are:

1. To select the type and speed of the runner for a given head. (See Art. 4–1.)
2. To estimate the maximum turbine efficiency. The maximum efficiencies which may be attained within a range of 2 per cent for specific speeds indicated are as follows:

* Lewis F. Moody, *Trans. A.S.C.E.*, Vol. 89 (1926), p. 628. The assumption of constant efficiency is satisfactory for preliminary calculations but cannot be used for refined design practice.

Type of Turbine	*Specific Speed*	*Efficiency*
Impulse	2–4	85–90
	4	90
	4–7	90–82
Francis	10–30	90–94
	30–82	94
	82–98	94–93
Propeller	100–140	94
	140–200	94–85

3. To determine the turbine setting for the reaction type of runner. (See Art. 4–9.)

2–14. Specific Diameter and Model Ratio. Eliminating Q from the following relations, $P \propto QH$ and $Q \propto D_1^2H^{1/2}$, in Eq. (2–18),

$$P \propto D_1^2H^{3/2} \tag{2–22}$$

solving for D_1,

$$D_1 \propto \frac{P^{1/2}}{H^{3/4}}$$

or

$$D_1 = D_s\frac{\sqrt{P}}{H^{3/4}} \tag{2–23}$$

It is apparent that the constant of proportionality D_s, to be called the specific diameter, is the diameter of a homologous runner developing 1 hp under 1-ft head. Let

$$m = \frac{\sqrt{P}}{H^{3/4}} \tag{2–24}$$

in which m is known as the model ratio, then

$$D_1 = mD_s \tag{2–25}$$

This means that the specific diameter multiplied by the model ratio should give the nominal diameter of the runner.

2–15. Synchronous Speed. In most cases in a modern hydroelectric plant the turbine is directly connected to an alternating-current generator of a given frequency, such as 25 or 60 cycles per second. Every alternating-current generator must have an even number of poles, and preferably a number of poles divisible by 4. For the sake of synchronization, the turbine speed must conform to the speed of the generator. In other words, the following relation will hold:

$$N = \frac{120f}{p} \tag{2–26}$$

where N is the synchronous speed, the rpm of the turbine and generator; f is the electrical frequency, in cycles per second, and p is the

number of generator field poles. The speed determined from theoretical considerations should be adjusted to conform with the nearest even number of poles, which number is divisible by four.

2–16. Turbine Parts and Auxiliaries. The vital part of a turbine is the runner, which has been discussed briefly in the preceding articles. For modern reaction turbines, the other essential parts are as follows:

The *stay ring* (Fig. 2–5) is a single casting consisting of upper and lower rings held together by vertical ribs known as *stay vanes,* or stationary guide vanes. The stay vanes are shaped to conform the flow

FIG. 2–5. Stay ring. (Allis-Chalmers Manufacturing Co.)

into the wicket gates, and are used to carry the load from above and take the force of the internal water pressure. The stay ring, provided only for concrete or steel spiral flumes and vertical units, forms a frame outside the wicket gates, to which the pit liner, cover plate, discharge

ring, and wicket gate rings are generally bolted (Fig. 2–2C). The stay ring is often called the speed ring.

The *wicket gates*, or movable guide vanes (Figs. 2–6, and 2–2C), are movable gates or vanes arranged on pivots, so that the degree of opening can be adjusted to properly control the flow of water into the runner. The wicket gates are operated by a gate-operating ring, which is coupled to the governor servometer.

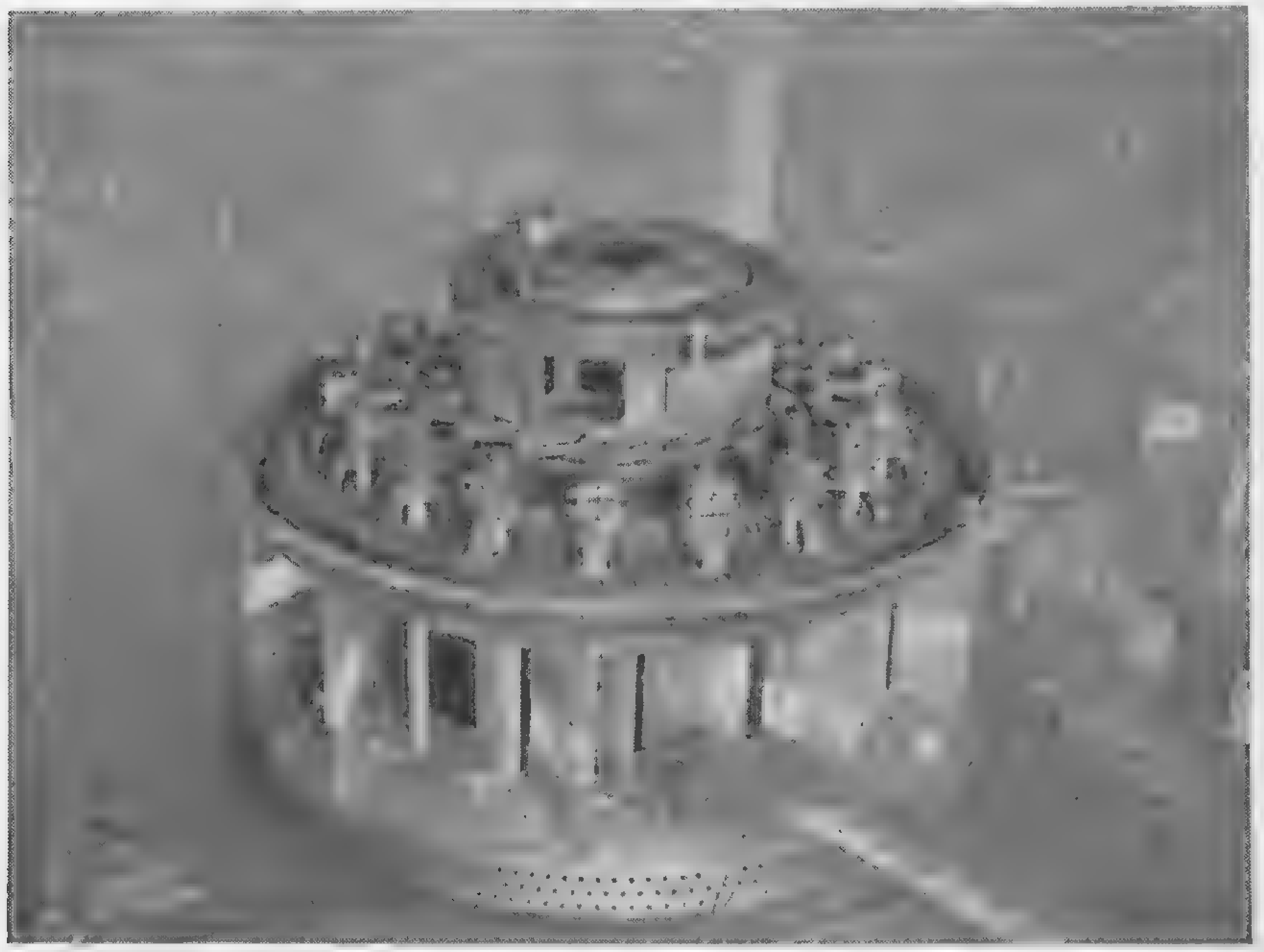


Fig. 2–6. Typical Francis turbine wheel case with adjustable wicket gates open. (S. Morgan Smith Co.)

The *distributor* is the assembly of wicket gates, together with the stay ring, head cover, and bottom ring.

The *governor* is a mechanical device designed to regulate the speed of the runner under fluctuating conditions of load in such a way as to control the amount of water delivered to the turbine in proportion to the load demand on the unit. If the turbine is of the Francis type or fixed-blade-propeller type, the governor opens or closes the wicket gates to a desired position. In the Kaplan movable-blade-propeller type, the governor has the additional function of adjusting the angle of the runner blades. (See Art. 6–14.)

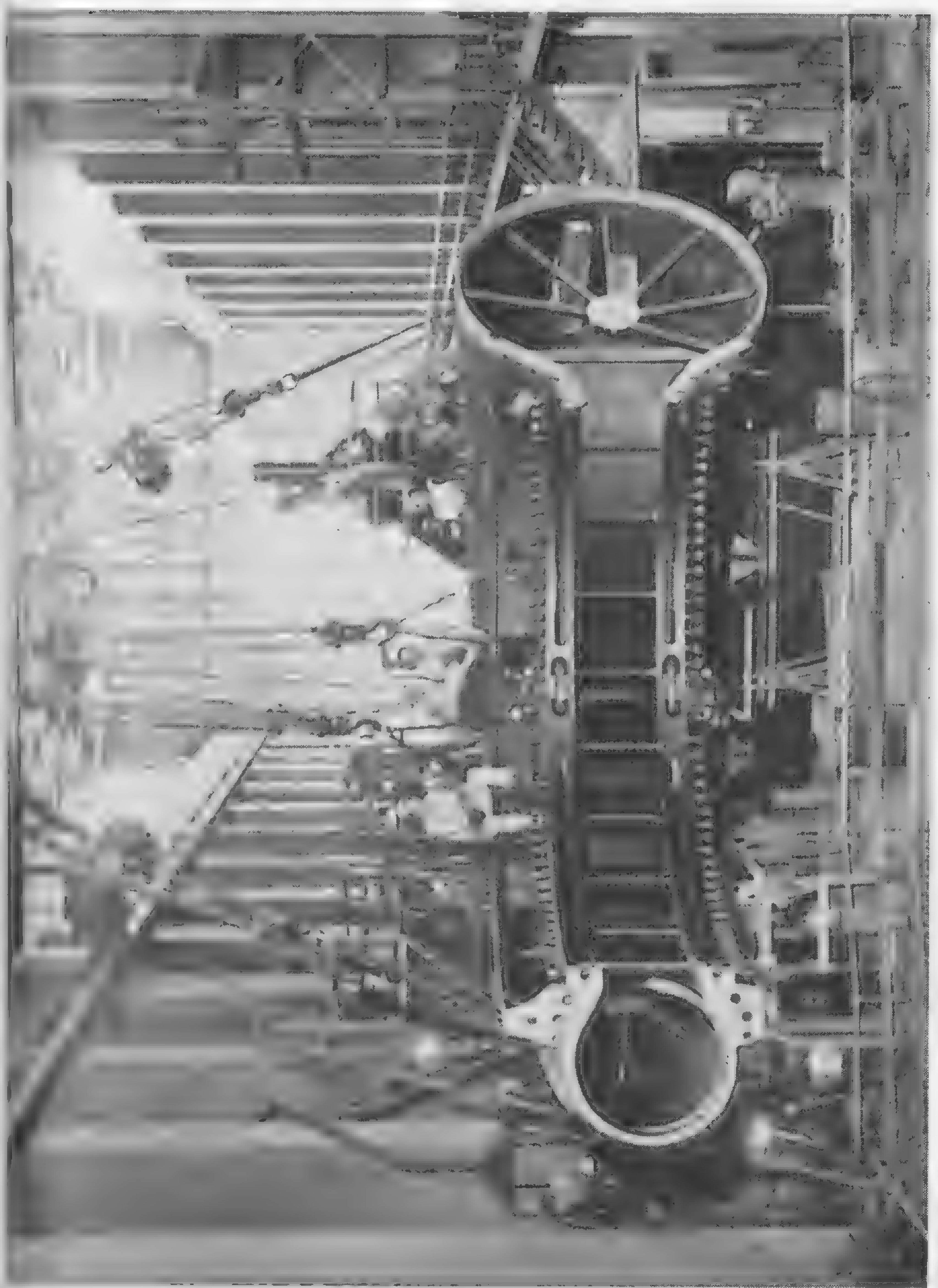


Fig. 2–7. Section of scroll case being lowered into Unit No. 1 pit, Shasta Dam. The illustration shows parts of the stay ring, guide vanes, and wicket gate openings. (Bureau of Reclamation)

Reaction turbines are installed either vertically or horizontally. They may be set in an open flume or enclosed in a casing such that water can be guided to the distributor and then to the runner.

2–17. Schemes of Water Power Development. Water power installations usually involve the construction of a dam. The dam may serve only to provide diversion or pondage, or it may serve to provide a large volume of storage. The powerhouse may be located immediately below the dam or at some distance from it. In most recent developments, the dam is a multi-purpose structure because it serves to create values other than power alone. Hydro power development is often only incidental to the primary purposes. However, since a dam raises the elevation of the water above it, head is created, and since flow is also involved, the two essential elements of water power are created—head and flow. Dams built primarily for the production of power may also serve secondary purposes such as municipal water supply and flood regulation.

Figure 2–8 shows the Kentucky Dam of the TVA, which serves to develop power, navigation, flood regulation, and recreational facilities. In this project the powerhouse is located directly below the dam. The pond above serves as the *forebay* and provides *pondage* for variations in hourly, daily, or weekly flows. The river below serves as the *tailrace*. Water is supplied directly to the turbines through concrete-lined passages in the dam. The lock at the left serves to lift or lower the river commerce.

Figure 2–9 shows a view of the Shasta Dam built by the Bureau of Reclamation. The primary purpose of this structure is to provide storage for irrigation water supply. Water is carried in steel *penstocks* from the storage reservoir. Power is an essential but not the primary component of this development. Other functions are flood regulation and recreation.

Figure 2–10 shows a layout of a plant removed from the dam in order to take advantage of the fall in the river from the base of the dam to the powerhouse site. The water is carried from the reservoir through a tunnel and a steel penstock provided with a surge tank which regulates water hammer pressures. *Intake* structures must be provided to admit water into penstocks, tunnels, canals, or flumes.

The term *headrace* is sometimes used to designate the channels used to conduct the water to the powerhouse. The *tailrace* is the channel used to conduct the water away from the powerhouse.

The *powerhouse* is the structure which houses and supports the weight of the machinery and facilities. The powerhouse may be an indoor or outdoor type. The indoor type consists of a building which completely

FIG. 2–8. Kentucky Dam on the Tennessee River. (Tennessee Valley Authority)

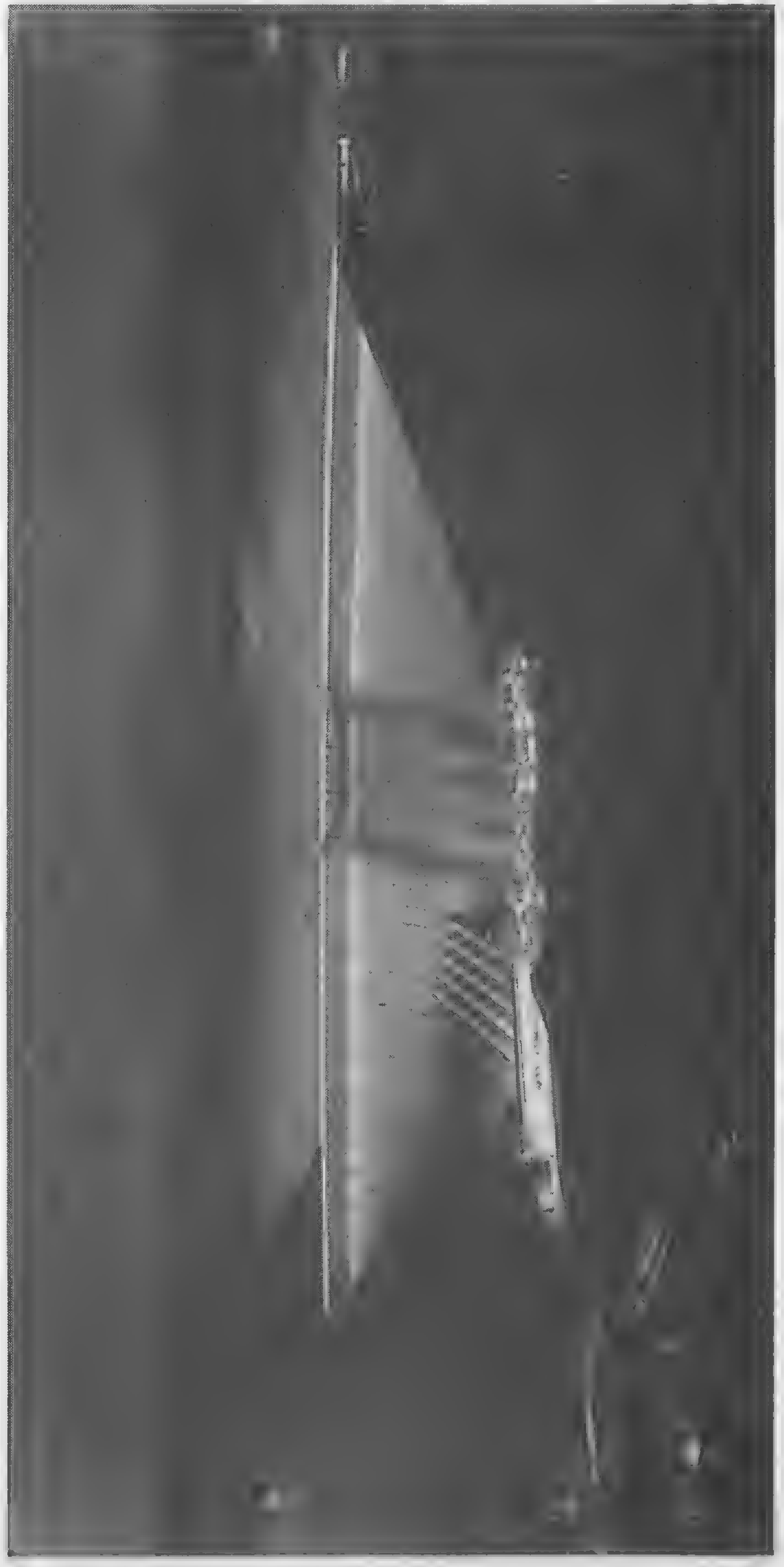


FIG. 2–9. Shasta Dam on the Sacramento River, California. (Bureau of Reclamation)

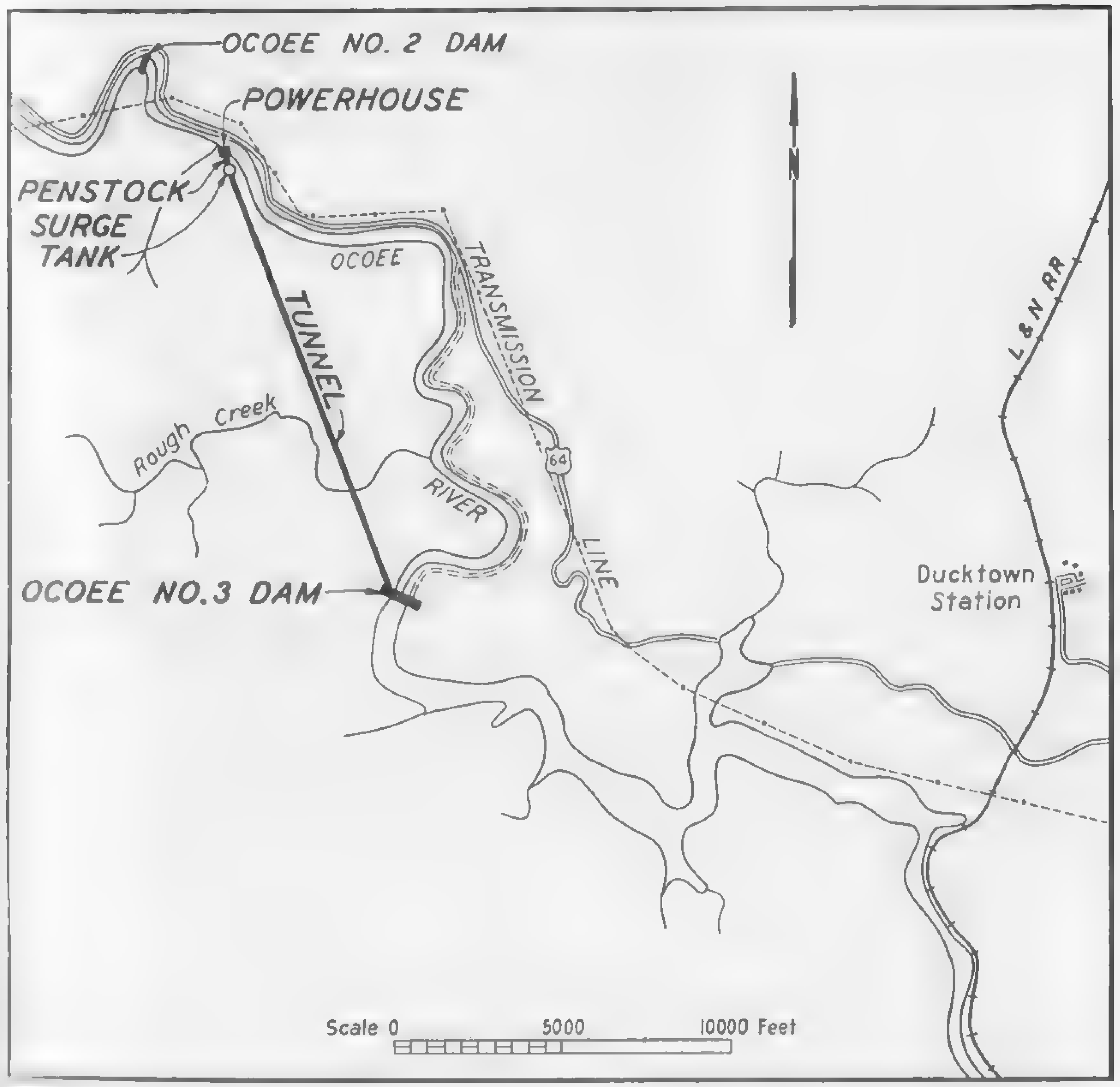


FIG. 2–10. Layout of Ocoee No. 3 on the Ocoee River, Tennessee. (Tennessee Valley Authority)

encloses the turbines, generators, and appurtenances. An overhead crane serves to lift the machinery (Fig. 2–11). The outdoor type consists of steel cases for the protection of generators. The control room and office spaces are enclosed. Figure 2–12 shows a view of this type of powerhouse. The two traveling gantry cranes serve the powerhouse and the transformers.

The layout of a water power development depends largely on the site characteristics, namely, the topographical features of the river and its vicinity, the head available, and the quantity of water to be used. The object is to utilize the maximum possible head and quantity of water consistent with cost, safety against flooding, provisions for droughts, and stability of foundation. In general, it is necessary to draw up several plans for alternative layouts before the most suitable arrangement is determined. Because site characteristics vary greatly, the lay-

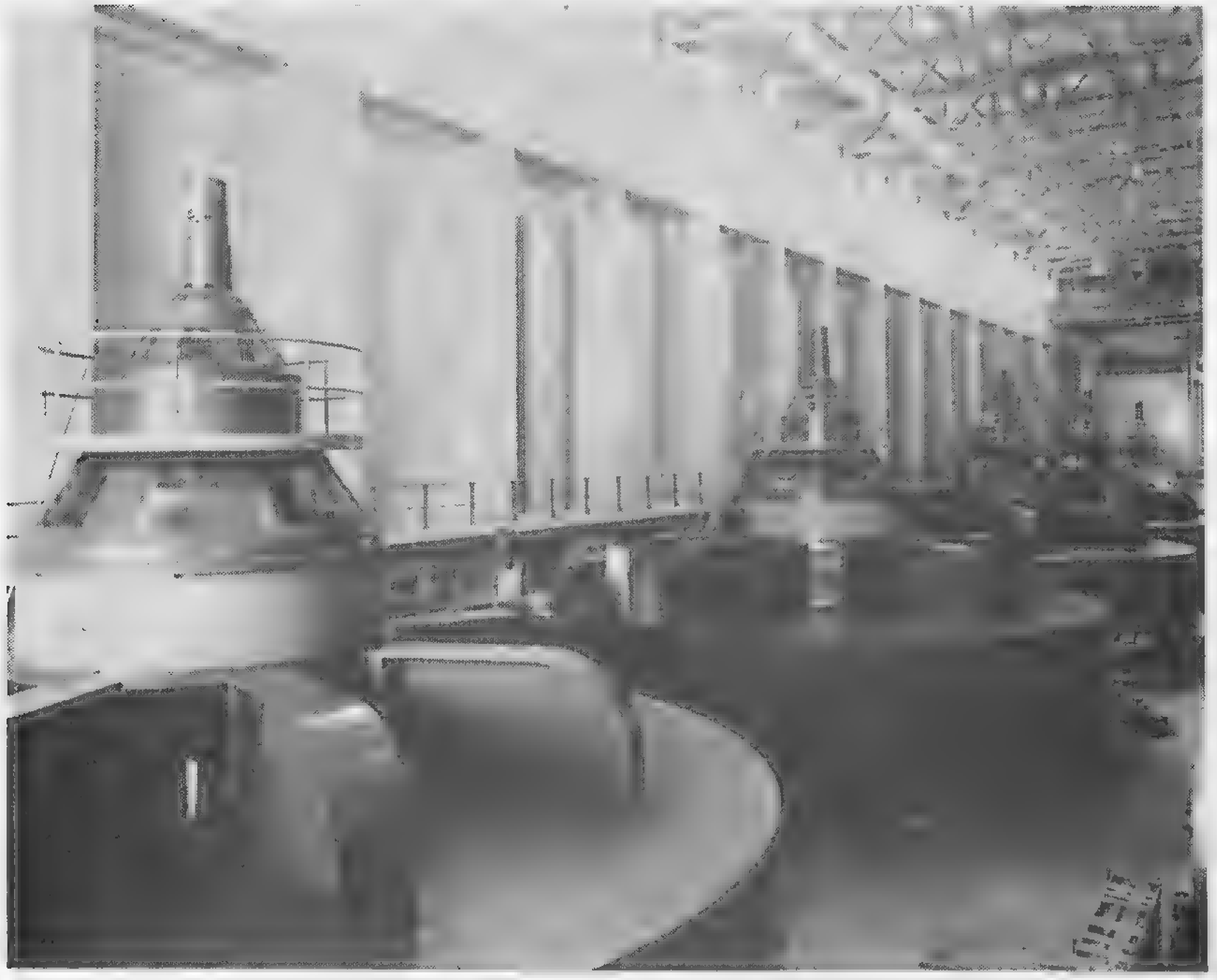


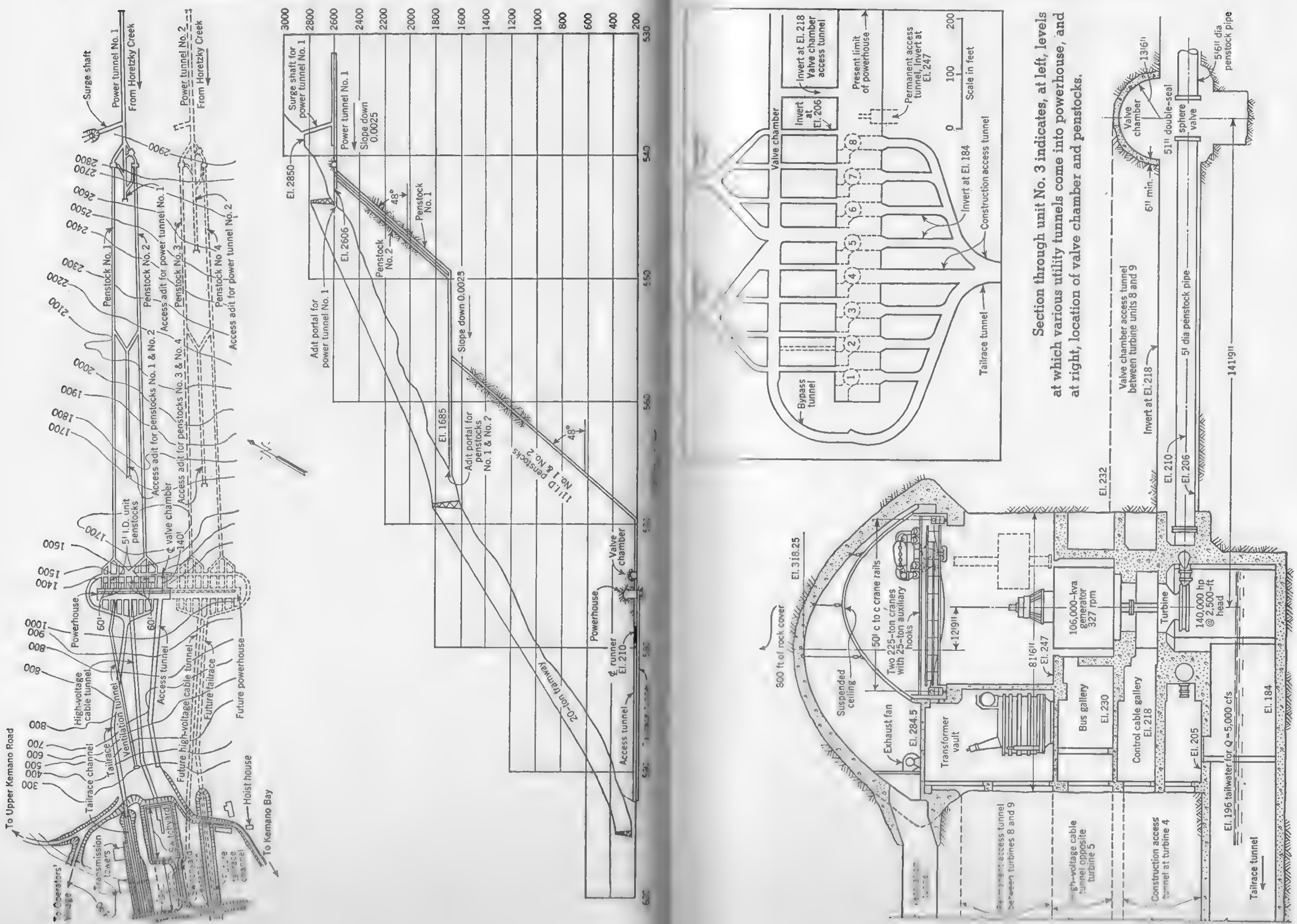
FIG. 2-11. Interior of Shasta powerhouse. (Bureau of Reclamation)

out of any particular development requires individual study and consideration. In general the type shown in Fig. 2-8 is adaptable to low-head developments; those shown in Figs. 2-9 and 2-10 are suitable for high-head projects.

2-18. Underground Hydro Power Stations. Underground hydroelectric plants have been extensively used in Europe, and considerable interest in them has recently developed among American engineers. This interest might be motivated by possible military advantages; lower costs resulting from improvements in tunneling machinery and methods; and the spiraling costs of above-ground steel penstocks, with expensive anchors, supports, and surge tanks. One underground plant at Snoqualmie Falls, Washington, is in operation in the United States. It has an installed capacity of 20,000 kw. Several such plants are being constructed in South and Central America.

A record-size underground plant, constructed for the Aluminum Company of Canada Ltd., was scheduled for completion in 1954. It is located in central British Columbia. A tunnel and penstock system about 11 miles long will conduct water from the Nechako River on the

FIG. 2-12. Wheeler Dam powerhouse. (Tennessee Valley Authority)



Section through unit No. 3 indicates, at left, levels at which various utility tunnels come into powerhouse, and at right, location of valve chamber and penstocks.

Fig. 2-13. Details of the Kemano (British Columbia) Development. (*Civil Engineering*, February and June, 1953)

eastern slope through the Coast Range to an underground powerhouse in the Kemano River Valley which drains to the Pacific Ocean. The ultimate installation will include 16 units of 140,000 hp, each located in an underground chamber 982 ft long, 81.5 ft wide, and 139 ft high from the bottom of the turbine pits to the crown of the roof arch. A center section 160 ft long and 118 ft high will serve as a service and control bay. The units are of the impulse-wheel type, operating at a rated head of 2500 ft. Each unit will require about 560 cfs to produce the rated horsepower. The top of Fig. 2–13 shows a plan and profile of the penstock and powerhouse section, and the bottom shows a cross section of the powerhouse.

CHAPTER 3

ANALYSES OF DATA

3–1. Stream-Flow Data. The stream-flow data required for the study of a hydro power development should include (1) daily, weekly, or monthly flow over a period of years; (2) minimum flow; and (3) flood, or maximum, flow.

The daily, weekly, or monthly flow data are used to construct the flow-duration curve. This curve is taken as a basis for the determination of plant capacity and power supply at all times. To accomplish this purpose, the periods of record from which the data are obtained should be of reasonable length, covering at least ten years or more. However, the reader is cautioned as to the reliability of some data, since records, particularly those secured from unofficial sources, are subject to errors of various kinds. Whenever such errors occur, the data should be corrected if possible, or rejected if the errors are large and corrections for them are impracticable. Furthermore, for the purpose of storage study, the data should include one or more dry periods of years. A mass curve based on dry-period data is necessary for the study of storage effect and flow regulation. A sample tabulation of the stream-flow data of a river is shown in Table 3–1, which lists the mean weekly discharges in cubic feet per second over a period of 20 years.

The minimum flow of a stream without storage regulation is the base flow due largely to the ground water contribution. It is usually the low-water flow or dry-weather flow of rivers, and is equal to zero for non-perennial rivers. Information including the magnitude and period of occurrence of the minimum flow is essential to the determination of the firm power (power available 100 per cent of time) and the design of plant auxiliaries. When the minimum flow is very low, it may be increased to a practical amount by means of artificial storage.

The flood, or maximum, flow of a stream is required for the adequate design of safety provisions, such as the spillway or gate relief. Safety provisions are needed to prevent damage or destruction to life and property near the region of development.

The collection of stream-flow data requires long-term and uninterrupted observation and recording. This work is usually undertaken by government agencies or private interested parties. In the United States, the Geological Survey is the principal federal agency engaged in the

TABLE 3–1

MEAN WEEKLY DISCHARGE, IN CFS

Week Ending	Week	1932	1933	1934	1935	1936	1937	1938	1939	1940	1941
Jan. 7	1	5,810	4,640	4,210	3,730	3,080	6,500	10,200	4,550	4,130	9,210
Jan. 14	2	3,890	4,060	4,610	3,200	2,960	6,780	8,040	3,850	7,100	10,400
Jan. 21	3	3,650	3,560	5,190	3,830	2,490	4,910	5,610	2,030	3,890	13,600
Jan. 28	4	3,130	3,580	5,950	3,920	2,040	3,890	5,830	1,710	5,700	6,750
Feb. 4	5	4,300	3,440	4,150	4,060	1,790	10,400	5,400	1,520	3,450	8,210
Feb. 11	6	4,040	4,500	5,680	5,030	2,100	7,100	7,750	1,510	3,040	9,770
Feb. 18	7	5,600	3,770	4,400	3,750	3,540	7,340	9,850	1,470	5,490	7,390
Feb. 25	8	7,160	3,520	5,030	3,400	3,040	7,640	8,960	1,460	3,630	6,420
Mar. 4	9	5,680	2,940	11,100	3,360	2,580	4,920	5,580	12,400	4,330	5,430
Mar. 11	10	10,000	4,330	12,100	6,660	2,500	4,230	5,080	9,120	5,060	4,680
Mar. 18	11	7,520	4,540	14,500	4,740	2,250	3,490	5,130	3,710	9,360	4,290
Mar. 25	12	5,270	4,100	10,800	6,490	2,440	5,420	6,960	4,790	4,710	6,870
Apr. 1	13	4,430	7,840	7,800	4,180	4,970	7,170	4,850	7,510	4,950	13,100
Apr. 8	14	4,350	5,790	5,670	3,500	8,600	6,090	4,440	3,820	6,840	21,600
Apr. 15	15	4,280	6,330	5,250	3,590	5,080	4,230	3,940	3,560	5,600	12,700
Apr. 22	16	3,580	5,050	4,880	3,400	5,300	3,420	6,700	3,660	5,400	7,000
Apr. 29	17	3,860	5,810	6,410	2,830	6,330	4,650	4,310	3,110	5,240	5,340
May 6	18	3,240	5,590	8,660	2,680	4,030	7,610	6,170	2,890	3,760	4,210
May 13	19	2,810	8,300	9,560	4,390	4,040	4,300	8,250	2,580	4,140	3,700
May 20	20	2,370	5,320	8,130	7,430	3,020	3,570	5,030	2,810	4,130	3,170
May 27	21	2,670	5,950	8,420	4,670	2,770	4,300	4,160	2,410	3,360	2,600
June 3	22	4,230	4,900	6,240	3,060	2,440	3,510	3,460	2,340	2,710	2,210
June 10	23	3,180	6,670	5,140	3,160	1,980	2,840	2,640	4,120	2,630	2,270
June 17	24	4,120	5,570	4,040	2,630	2,030	3,480	2,390	3,580	2,300	2,210
June 24	25	5,010	4,110	3,780	2,330	1,710	2,800	2,020	3,750	2,260	1,670
July 1	26	3,210	6,140	6,040	2,100	1,830	2,390	2,140	2,520	1,880	1,410
July 8	27	2,810	4,860	3,960	1,580	2,090	4,100	1,850	2,440	1,960	1,890
July 15	28	2,780	5,110	3,860	1,670	1,690	2,450	1,800	2,530	1,560	1,590
July 22	29	2,830	3,520	3,580	1,840	2,450	2,040	1,950	2,020	1,940	1,900
July 29	30	2,660	3,730	3,070	1,440	2,280	1,660	2,040	2,000	2,820	1,240
Aug. 5	31	2,500	3,320	3,250	1,260	2,050	2,100	2,080	1,820	1,670	2,180
Aug. 12	32	3,160	3,640	2,400	1,340	1,790	1,910	1,610	2,020	1,570	1,620
Aug. 19	33	3,250	9,940	2,280	1,070	1,590	2,440	1,670	2,930	1,710	1,500
Aug. 26	34	2,170	5,140	1,910	1,420	1,840	1,760	1,440	2,600	4,060	1,370
Sept. 2	35	2,160	4,840	1,620	943	1,710	1,290	1,170	2,150	1,910	1,640
Sept. 9	36	2,380	11,200	1,770	1,080	2,210	1,190	990	1,600	1,750	1,530
Sept. 16	37	1,970	5,080	1,850	1,890	1,240	981	1,880	1,560	1,430	1,190
Sept. 23	38	1,900	3,890	2,670	1,960	1,130	1,220	1,320	1,440	1,160	1,130
Sept. 30	39	1,770	2,800	8,250	1,590	1,070	1,720	1,100	1,720	1,070	3,330
Oct. 7	40	1,070	2,490	5,430	1,080	885	2,310	1,180	3,130	930	3,040
Oct. 14	41	2,570	2,150	2,900	1,170	903	1,470	970	3,500	920	3,750
Oct. 21	42	1,480	4,150	2,260	946	761	5,630	1,040	2,060	950	3,580
Oct. 28	43	1,190	3,990	2,640	991	805	3,040	960	1,600	970	2,020
Nov. 4	44	1,360	2,700	5,320	1,120	1,040	4,190	990	2,730	1,430	2,120
Nov. 11	45	1,350	2,340	5,240	986	805	3,040	1,060	2,590	1,320	2,470
Nov. 18	46	2,980	2,130	9,630	3,580	802	2,490	1,330	1,900	4,490	2,290
Nov. 25	47	2,320	3,300	6,170	2,090	1,030	3,910	1,100	3,100	1,750	1,790
Dec. 2	48	2,220	2,420	4,960	1,580	1,410	3,810	870	4,950	1,920	1,640
Dec. 9	49	6,300	2,290	4,200	2,520	4,370	2,680	1,320	4,600	1,620	3,670
Dec. 16	50	6,990	2,230	3,410	1,750	7,680	10,700	1,100	2,840	2,300	2,890
Dec. 23	51	6,490	2,530	3,890	1,600	5,310	9,210	1,300	2,910	1,690	4,240
Dec. 31	52	4,590	2,160	3,600	2,340	4,450	10,000	1,420	3,020	1,420	5,250
Mean for year, cfs		3,660	4,467	5,343	2,749	2,060	4,411	3,470	3,130	3,100	4,570

TABLE 3–1 (*Continued*)

MEAN WEEKLY DISCHARGE, IN CFS

Week Ending	Week	1942	1943	1944	1945	1946	1947	1948	1949	1950	1951
Jan. 7	1	16,400	4,710	2,850	907	3,850	3,810	6,940	3,780	5,870	10,500
Jan. 14	2	7,370	3,030	2,920	1,550	2,430	2,410	4,280	2,520	3,380	16,700
Jan. 21	3	12,400	2,550	3,200	2,210	2,560	2,540	7,320	2,790	3,090	6,560
Jan. 28	4	10,300	3,870	3,580	1,090	3,040	2,380	7,310	2,090	2,580	4,490
Feb. 4	5	7,690	3,610	10,500	1,120	2,510	2,940	7,720	1,900	2,070	6,800
Feb. 11	6	8,340	2,930	10,300	2,420	2,000	4,990	12,700	3,870	2,410	14,500
Feb. 18	7	6,590	2,860	16,700	3,090	2,140	6,450	5,950	8,910	9,060	9,900
Feb. 25	8	6,220	3,540	7,310	4,580	1,730	4,830	4,570	8,940	8,040	6,470
Mar. 4	9	4,760	3,250	9,340	4,060	1,840	3,250	3,670	12,800	5,570	4,720
Mar. 11	10	4,240	5,740	11,300	3,160	3,660	6,290	3,840	7,310	4,870	7,630
Mar. 18	11	4,130	5,770	6,610	4,620	3,400	7,970	6,490	5,130	3,510	8,650
Mar. 25	12	4,010	5,590	4,750	3,330	2,900	7,780	10,800	10,200	4,050	6,340
Apr. 1	13	3,630	5,300	5,190	3,490	3,290	5,020	6,510	13,500	5,400	7,980
Apr. 8	14	4,060	6,370	5,250	3,260	3,660	3,600	4,570	7,160	4,390	5,470
Apr. 15	15	4,420	6,730	4,450	3,230	2,960	3,430	4,240	7,180	3,100	5,320
Apr. 22	16	3,620	5,350	4,070	5,990	2,390	2,670	5,760	6,400	5,120	6,330
Apr. 29	17	5,950	4,500	4,370	4,820	2,950	2,400	5,820	6,200	6,740	5,090
May 6	18	5,100	3,570	3,590	3,390	2,270	2,460	4,050	4,850	6,000	6,770
May 13	19	3,960	3,180	3,110	3,000	2,000	2,050	4,290	4,150	5,270	5,550
May 20	20	3,570	3,660	2,870	2,490	1,690	3,200	3,830	3,590	4,980	7,520
May 27	21	2,910	4,360	2,940	2,270	1,450	8,350	4,330	3,220	3,610	5,130
June 3	22	2,920	5,320	3,290	1,960	1,460	3,860	3,490	3,170	2,690	4,150
June 10	23	2,790	3,930	2,840	2,070	1,400	3,810	3,020	2,800	2,910	3,320
June 17	24	2,530	3,210	2,550	2,900	1,900	4,210	3,090	2,420	2,410	3,460
June 24	25	2,600	3,820	2,270	2,560	1,310	2,700	2,530	1,890	2,520	2,600
July 1	26	1,960	3,430	2,100	2,170	1,460	2,280	3,310	1,660	1,880	2,490
July 8	27	2,200	2,770	2,120	2,430	6,230	2,300	7,310	1,690	1,790	2,760
July 15	28	1,900	2,490	2,840	2,560	4,450	5,670	4,900	1,940	1,760	2,740
July 22	29	1,850	4,820	1,800	2,580	4,820	2,580	3,370	1,710	1,800	2,780
July 29	30	2,110	10,500	1,840	1,900	3,020	3,170	3,120	1,360	2,260	2,140
Aug. 5	31	2,030	6,000	1,660	1,660	2,500	2,680	4,590	1,810	2,680	1,830
Aug. 12	32	2,550	5,290	1,510	1,540	2,490	3,130	2,960	1,530	2,660	1,710
Aug. 19	33	2,530	3,620	2,390	8,960	1,900	3,010	3,380	2,040	1,640	1,480
Aug. 26	34	2,940	2,670	1,850	2,670	1,850	4,130	2,170	1,180	1,840	1,470
Sept. 2	35	2,890	2,400	1,590	12,000	1,680	2,390	1,810	1,130	1,280	1,320
Sept. 9	36	3,150	2,480	1,390	4,260	1,500	2,960	1,810	1,130	1,160	1,130
Sept. 16	37	2,640	2,380	1,080	2,830	1,270	2,330	1,470	1,300	3,490	1,350
Sept. 23	38	1,840	1,780	1,080	2,200	1,040	1,930	2,710	1,340	3,350	1,330
Sept. 30	39	1,550	1,640	1,010	1,950	1,090	3,530	1,640	2,290	1,680	1,350
Oct. 7	40	1,930	1,480	1,040	1,660	900	2,430	1,470	1,870	2,160	865
Oct. 14	41	1,590	1,330	868	1,570	914	1,940	1,380	1,310	1,510	2,010
Oct. 21	42	5,010	1,290	826	1,470	847	1,620	1,670	1,420	1,410	1,070
Oct. 28	43	3,970	1,260	909	1,310	1,330	2,200	1,510	1,310	2,800	1,610
Nov. 4	44	2,970	1,200	918	2,160	1,750	2,290	1,500	1,190	1,740	1,220
Nov. 11	45	2,260	3,050	816	1,620	1,330	1,880	2,230	1,270	1,480	1,580
Nov. 18	46	2,130	1,570	826	2,110	1,050	1,600	1,570	1,260	2,230	1,850
Nov. 25	47	1,830	3,170	992	1,490	1,780	1,670	1,380	1,580	4,240	1,650
Dec. 2	48	2,220	1,850	940	1,860	1,360	2,570	1,310	2,870	2,850	1,740
Dec. 9	49	1,780	1,690	904	1,570	2,890	6,650	1,450	2,160	4,980	1,230
Dec. 16	50	1,820	1,780	814	1,750	2,080	3,940	1,360	1,880	3,730	1,300
Dec. 23	51	2,810	1,540	1,020	2,190	1,920	2,940	1,190	1,910	2,690	2,980
Dec. 31	52	3,770	2,740	1,420	3,420	3,830	14,200	2,380	5,160	6,080	3,210
Mean for year, cfs		3,080	3,520	3,310	2,840	2,270	3,710	3,880	3,540	3,400	4,230

collection and publication of stream-flow data. Data from about 5000 stations are now published in annual Water Supply Papers by the Survey. Other government agencies such as the Bureau of Reclamation, Corps of Engineers, Soil Conservation Service, International Boundary Commission, and Tennessee Valley Authority also collect stream-flow data for their related projects.

It often happens that the data available are not recorded at the site of the development. If data at a nearby station are available, it is possible to transpose the data to the site under consideration, at the expense of a certain amount of accuracy. The Geological Survey has begun a program of analyzing stream-flow data. One of the first publications based on these studies, "Water Supply Characteristics of Illinois Streams" by W. D. Mitchell, gives data and flow-duration curves at selected stations.* The flow data are given in second-feet per square mile, so that flow at any point on the stream, at any per cent of time, is readily determinable if the drainage area is known. Precipitation records are generally longer and more easily obtained than discharge records. Therefore, it is also possible to estimate the stream flows from available precipitation records. The student is urged to explore recent papers and books on the subject of supplying reliable missing flow data. The science of hydrology is advancing rapidly and increasingly more reliable contributions have been appearing in engineering literature.

3–2. Flow-Duration Curve. The flow-duration curve is obtained by plotting the values of stream flow, in order of magnitude, as ordinates, and plotting per cent of time as abscissas. The curve shows flow equaled or exceeded for any desired per cent of time covered by the record.

As an example, a flow-duration curve based on the mean weekly stream-flow data given in Table 3–1 is constructed in Fig. 3–1. From the resulting curve, the discharge equaled or exceeded 20 per cent of the time, for instance, is 5000 cfs. Computations involved in the construction of this curve are given in Table 3–2. This table shows only the beginning and end of the calculations, and is offered for illustrative purposes only. The first column of the table gives the discharge arranged in order of decreasing magnitude. The second column gives the number of times the discharge in Column 1 was equaled or exceeded. The third column is obtained by dividing the values in Column 2 by the total number of records in Column 1, and multiplying by 100 to convert to percentage. In this example, the number of records is 1040, or 20 years multiplied by 52 weeks. The flow-duration curve is obtained by plotting values in Column 1 against those in Column 3. Figure 3–1

* W. D. Mitchell, *Water Supply Characteristics of Illinois Streams*, Illinois Division of Waterways, in cooperation with the U. S. Geological Survey, 1950.

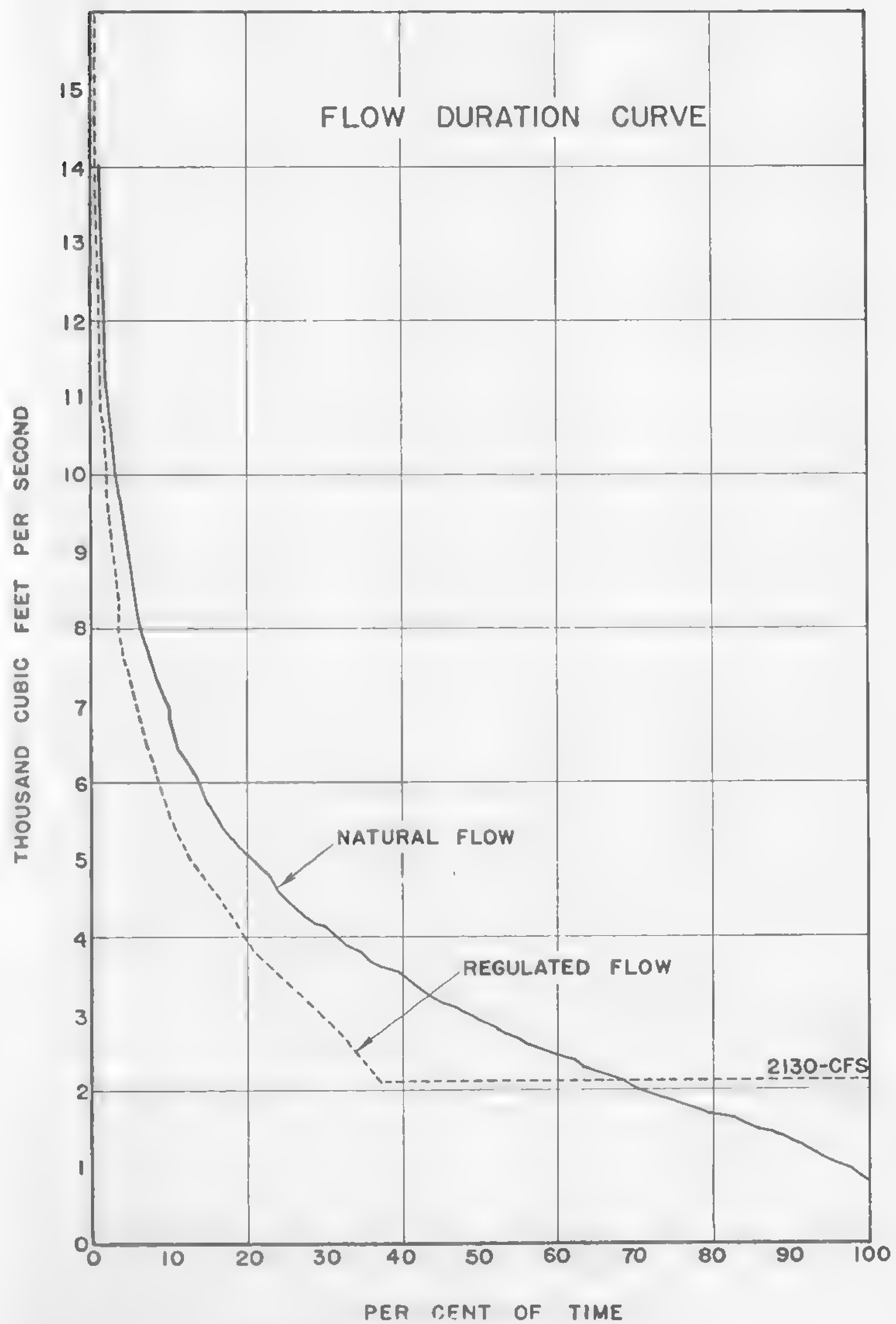


Fig. 3–1. A flow-duration curve.

TABLE 3–2

COMPUTATIONS FOR WEEKLY FLOW-DURATION CURVE
(Plotted in Fig. 3–1)

(1) Discharge cfs.	(2) Number of Times Discharge is Equaled or Exceeded	(3) Per Cent of Time
21,600	1	0.096
19,000	2	0.192
16,700	4	0.384
16,400	5	0.480
14,500	7	0.673
14,200	8	0.769
....	..	.....
....	..	.....
....	..	.....
847	1032	99.231
826	1033	99.327
816	1035	99.519
814	1036	99.615
805	1038	99.808
802	1039	99.904
761	1040	100.000

also shows the flow-duration curve if the flow is regulated by storage for a maximum draft of 2130 cfs, 100 per cent of the time. This is as used in the example given in Art. 3–10. The regulated flow-duration curve is adjusted for losses as explained in Art. 3–4.

3–3. Power-Duration Curve. When the available power in a stream is plotted against the per cent of time, a power-duration curve results. The available power in horsepower or kilowatts of a stream is computed by Eq. (2–2a) or (2–2b), for which data on discharge, head, and efficiency are required. Assuming a normal operating head of 50 ft and an average plant efficiency of 88 per cent, from Eq. (2–2b), the factor used to convert the discharge into power in kilowatts is $\frac{weH}{737} = \frac{62.4 \times 0.88 \times 50}{737}$, or 3.74. Then the flow-duration curve in Fig. 3–1 may be readily converted into a power-duration curve by multiplying the ordinates by a conversion factor, 3.74, resulting in the curve shown in Fig. 3–2. Similarly a horsepower-duration curve could be constructed by multiplying the ordinates by $\frac{62.4 \times 0.88 \times 50}{550} = 5.0$.

3–4. Mass Curve. The mass curve is constructed by plotting the cumulative values of stream flow against the time throughout the period of record. It should be noted that when the mass curve is to be used

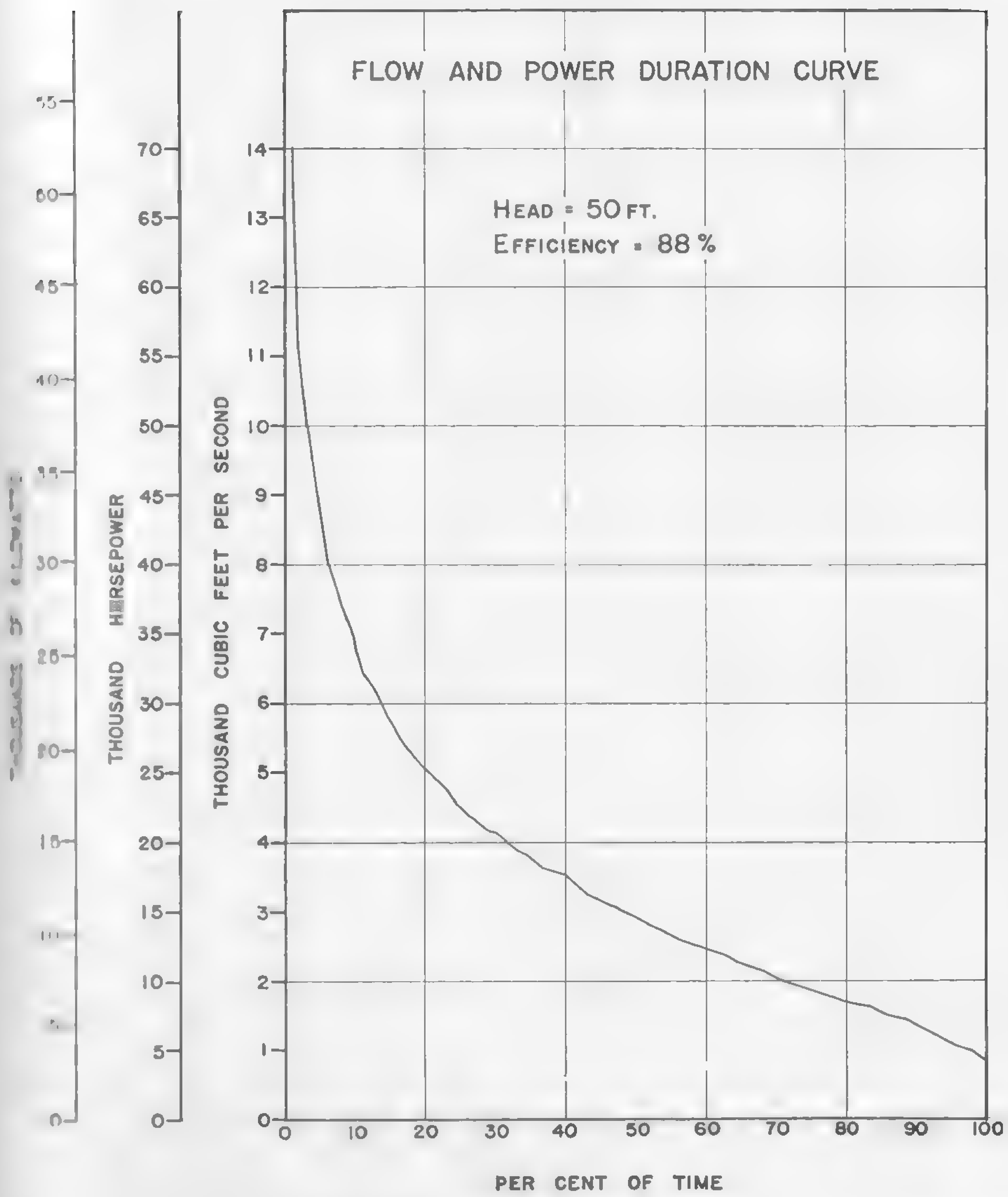


FIG. 3–2. Flow- and power-duration curve.

for estimating storage requirements and allowable draft for power purpose, the natural flow data must be adjusted to account for the following: (1) all losses due to evaporation and leakage from the proposed reservoir, and (2) all releases of water demanded by prior rights or purposes, but which for some good reason cannot be used for power generation. Such rights or purposes may include: demands by older water rights, municipal or industrial water supplies, irrigation, flood control, flow regulation for sanitary purposes, and navigation. The data for the illustrative mass curve shown in Fig. 3–3 were computed by adjusting the natural flow data, given in Table 3–1, for evaporation and leakage only. The adjustment was made on the following assumptions for the proposed reservoir: normal pool level elevation, 1710 ft above M.S.L.; area of reservoir at El. 1710, 10,700 acres; annual evaporation loss, 50 in., or 4.2 ft; leakage, 68 sec-ft weeks. (See Table 3–3.)

The total annual loss due to evaporation is therefore $10{,}700 \times 4.2 = 44{,}900$ acre-ft, an average of $44{,}900 \div 52 = 864$ acre-ft per week, or 432 sec-ft weeks. Total loss $= 432 + 68 = 500$ sec-ft weeks. The natural flow data given in Table 3–1 were reduced by 500 to determine the net available flow after construction of the reservoir.

The above-explained adjustment provides a rather rough correction, since it is based upon the annual evaporation. A more refined adjustment would take into account the monthly or weekly evaporation as it varied for different climatological conditions at the site of the reservoir. The mass curve as a device used for storage computation is discussed in Art. 3–10.

3–5. Firm Power. Firm power, or primary power, is theoretically the power which a hydroelectric plant may be depended upon to produce at all times. However, on reservoir-regulated streams it has become the practice to consider firm power as that which may be depended upon 95 per cent of the time. This practice would result in a shortage of power on an average of only 5 years in 100 years. From an economic point of view, it is relatively sound. For run-of-river plants, the flow available 97 per cent of the time is a safer criterion than 95 per cent. Firm power is not necessarily produced continuously. In cases where pondage is available and there is an interconnection, the hydro plant may be operated on the peak load only. However, the total power which can be produced by the hydro plant is limited by the total amount of water furnished by the stream during a short period of time, such as a day or a week. The total amount of primary power in kilowatt-hours or horsepower-hours is therefore indicated by the area of the duration curve under the 95 or 97 per cent ordinate, or under the amount of minimum flow as regulated by storage.

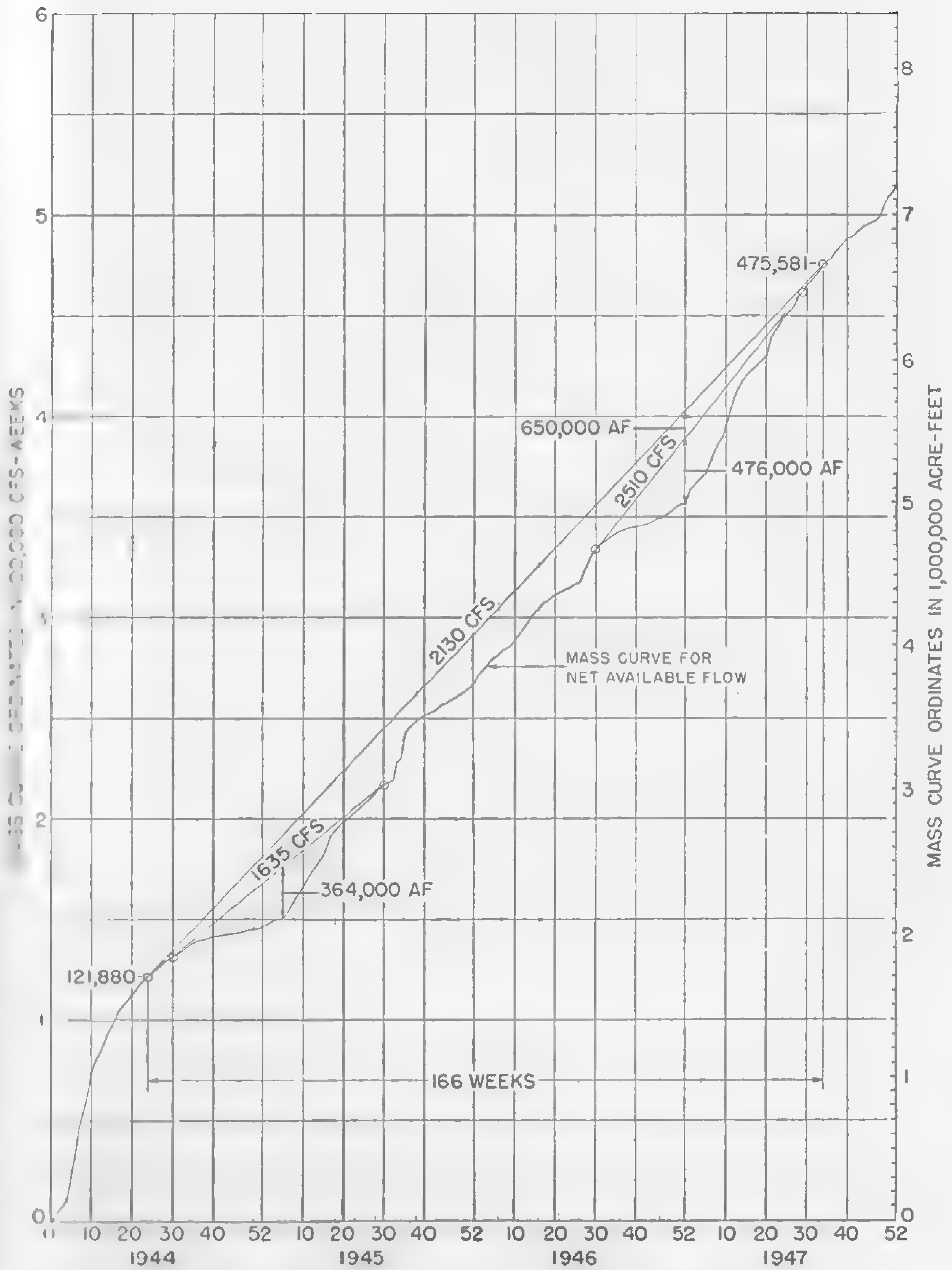


Fig. 3–3. Mass curve.

TABLE 3-3

CONSTRUCTIONS FOR MASS CURVE
(Plotted in Fig. 3-3)

Year (1)	Week of Year (2)	Week of Period (3)	Discharge, cfs (4)	Mass Curve Ordinates, week cfs (5)
1944	1	1	2,350	2,350
1944	2	2	2,420	4,770
1944	3	3	2,700	7,470
1944	4	4	3,080	10,550
1944	5	5	10,000	20,550
1944	6	6	9,800	30,350
1944	..	...	...	...
1944	..	...	...	...
1944	..	...	...	...
1944	24	24	2,050	121,880
....	..	...	...	...
....	..	...	...	...
....	..	...	...	...
1947	34	190	3,630	475,581
1947	..	...	...	...
1947	..	...	...	...
1947	..	...	...	...
1947	47	203	1,170	497,851
1947	48	204	2,070	499,921
1947	49	205	6,150	506,071
1947	50	206	3,440	509,511
1947	51	207	2,440	511,951
1947	52	208	13,700	525,651

The average flow for period $= 525{,}651/(4 \times 52 = 208) = 2530$ cfs per week. The greatest draft $= (475{,}581 - 121{,}880)/166 = 2130$ cfs per week. (See Art. 3-10.)

Note: Values given in this table may be checked from the data given in Table 3-1. The time of beginning of critical dry period (24th week of 1944) and lowest point of reservoir drawdown (34th week of 1947, 190th week of total period) were determined from the mass curve study, Fig. 3-3.

3-6. Surplus Power. Surplus, or secondary, power is all of the available power in excess of the firm power. The surplus power in kilowatt-hours is indicated by the area under the power duration curve between the horizontal line representing the firm power and the horizontal line representing the total installation in the power plant. Surplus power is sold at a cheaper rate than firm power because the customer is subject to a shutoff when the amount of power available is less than the amount required under firm contracts.

3-7. Load Curve. The load curve is plotted by using kilowatts as ordinates and time as abscissas, showing the variation of the load in a designated period, usually 24 hours or a week. A typical daily load

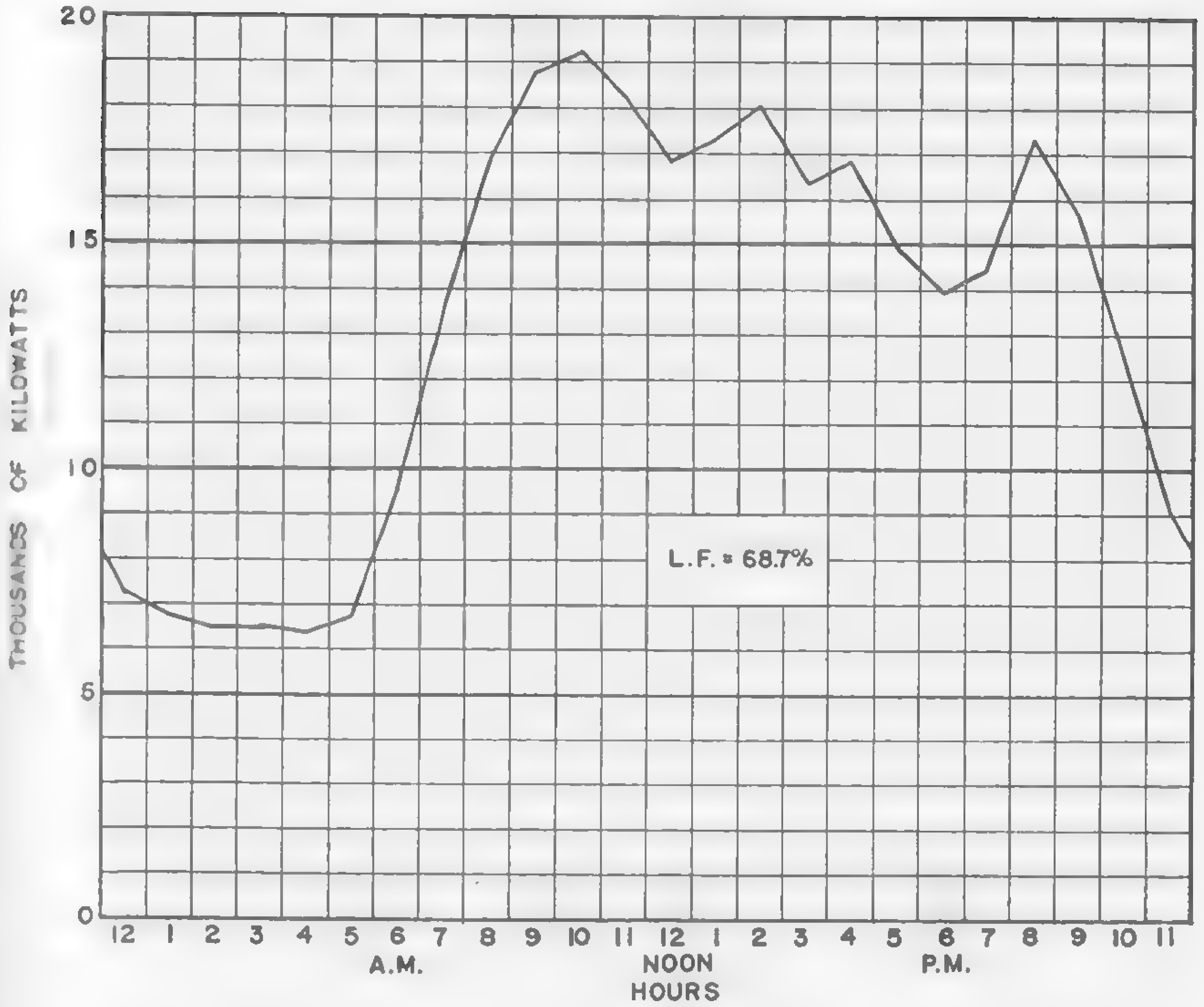


FIG. 3-4. Typical load curve for a summer day in an average community. (Illinois Power Co.)

curve is shown in Fig. 3-4. For convenience of analysis, the ordinate of the load curve may be expressed in per cent of the peak load. Such a load curve is called a *unit load curve.* The shape of the load curve varies mainly with the season of the year, the nature of the demand, and the work and play habits of consumers. Saturday, Sunday, and holiday loads are usually much lower than weekday loads. Industrial, municipal, domestic, and transportation demands for electric power all reflect the specific needs and habits of the industries and the people who make use of the power. General patterns are usually set up for a given area and may vary greatly in different areas.

The load curve for the peak day of the year is usually the most critical single-day load curve which is used for engineering study to determine the means of serving the loads. However, other days, weeks, and months should be investigated as necessary. The weekly load curve during a winter season, Fig. 3-5, is considered by some engineers to be the most useful for analytical purposes because the general pattern varies very little during the year. The objective of the power producer

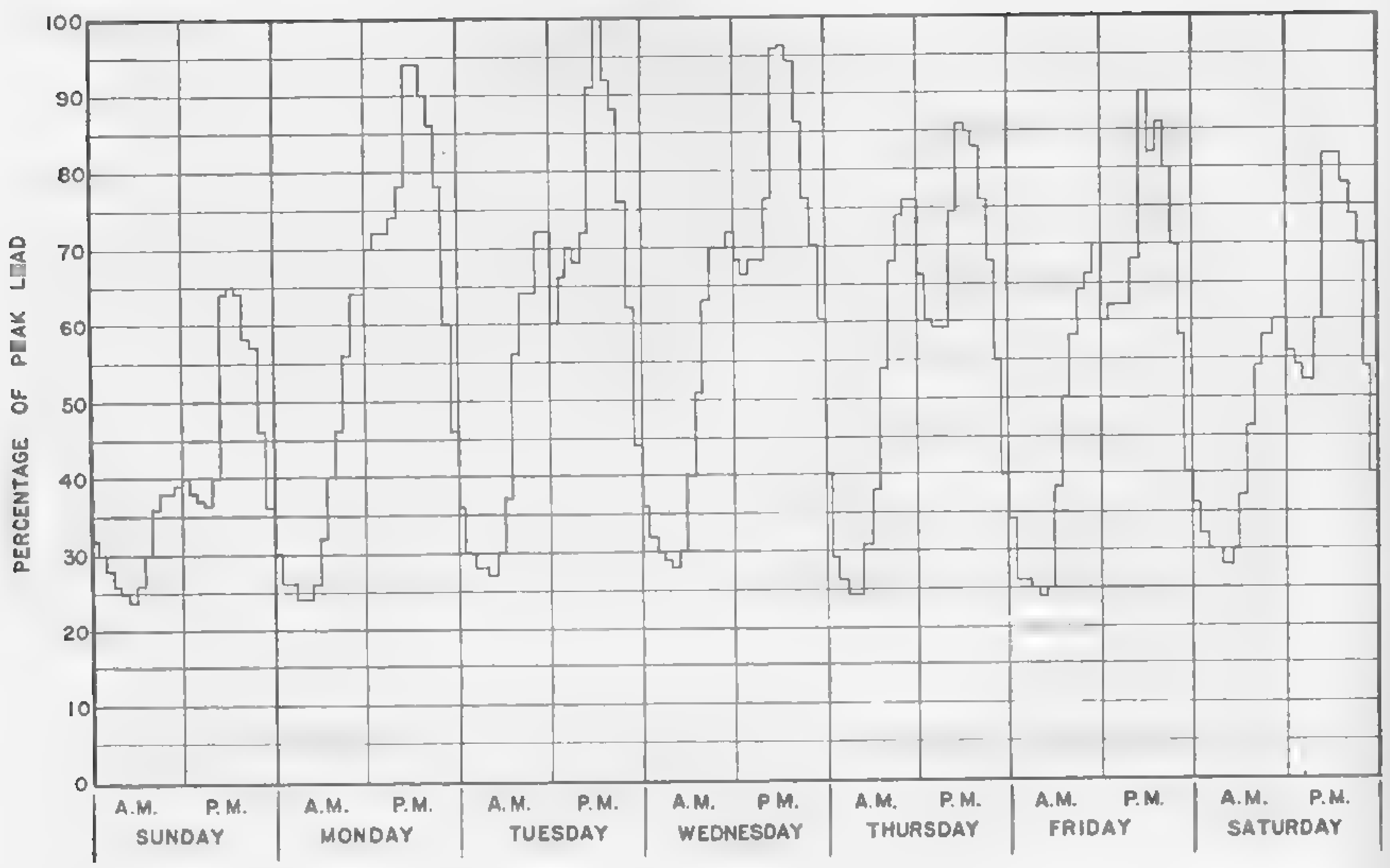


Fig. 3–5. Typical weekly unit load curve for a maximum winter week in an average community. (Illinois Power Co.)

is to fill up the valleys so as to improve the load factor. The typical peak-day unit load curves for industrial, domestic, and lighting uses, and the metropolitan system are shown in Fig. 3–6.

As shown in these load curves, the load of a power plant can be divided into two parts, the peak load and the base load. The peaks require the installation of extra units with capacities greatly exceeding those required for the base load. Because of the short duration of the peak loads, the units are operated at full capacity only for short periods. In case of hydro steam association, the hydro plant is operated to carry the base load during periods of high flow and the peak load during periods of low flow.

3–8. Load Factor. The load factor is defined as the ratio of the average load over a designated period to the peak load occurring within the same period. The period may be a day, a week, a month, or a year.

The load factor may be computed easily from the load curve. According to the definition given above, the load factor is equal to the area under the load curve divided by the product of the peak load and the designated period. For instance, the area under the load curve of Fig. 3–3 is 316,000 kw-hr, and the load factor is $(316{,}000)/(19{,}200 \times 24 \text{ hr}) = 68.7$ per cent.

The load factor varies greatly according to the character of the load. In general, a load system for an industrial region has an annual load

factor of 40 to 60 per cent, and for a metropolitan area without much manufacturing business the annual load factor is 30 to 35 per cent.

3–9. Pondage. In order to regulate the variable power demand, pondage is required so that excess water in the stream can be ponded to meet the deficiency of the supply. The computations for pondage may be illustrated by the following:

Illustrative Example: The peak load of an isolated plant is 45,000 kw, operated under an average head of 60 ft and at an average efficiency of 88 per cent. Using the daily unit load curve of Fig. 3–7, compute (1) the daily pondage required in acre-feet, and (2) the amount of spill or losses in acre-feet, that would occur on a day when the flow is 5000 cfs.

Solution: Computations are shown in Table 3–4, in which the first two columns give the hour and the hourly load demand in per cent of peak load as obtained from the given unit-load curve. The sum of the percentages in Column (2) is 960 per cent of peak hours, which gives a daily load factor of $960/24 = 40$ per cent.

From the given data, the discharge required for peak load is

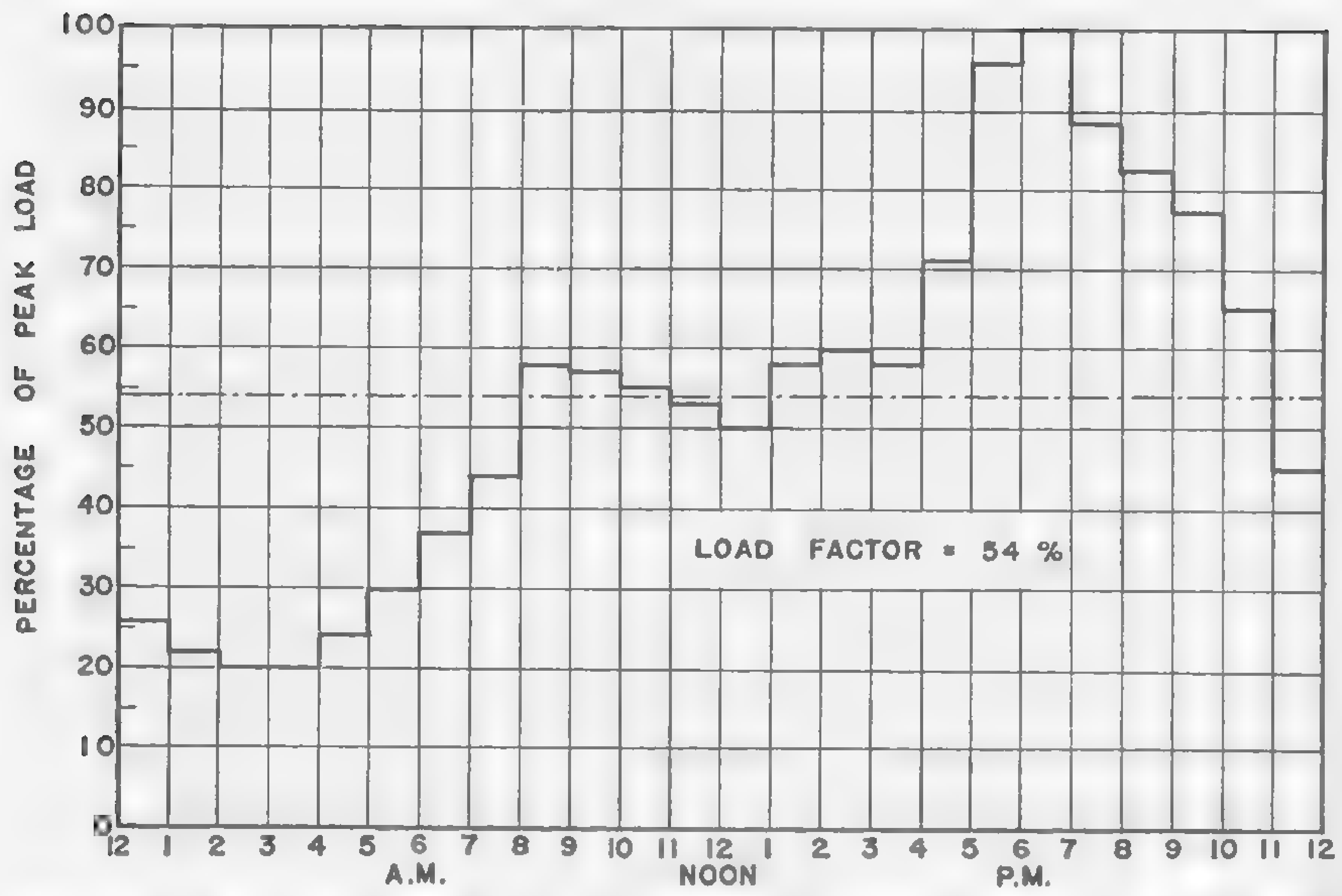
$$\frac{1.34 \times 45{,}000 \times 550}{62.5 \times 0.88 \times 60} = 10{,}050 \text{ cfs}$$

With a load factor of 40 per cent, the average discharge is equal to $10{,}050 \times 0.40 = 4020$ cfs. Column (3) gives the discharges for hourly demands, which were obtained by multiplying the percentages in Column (2) by 10,050 cfs.

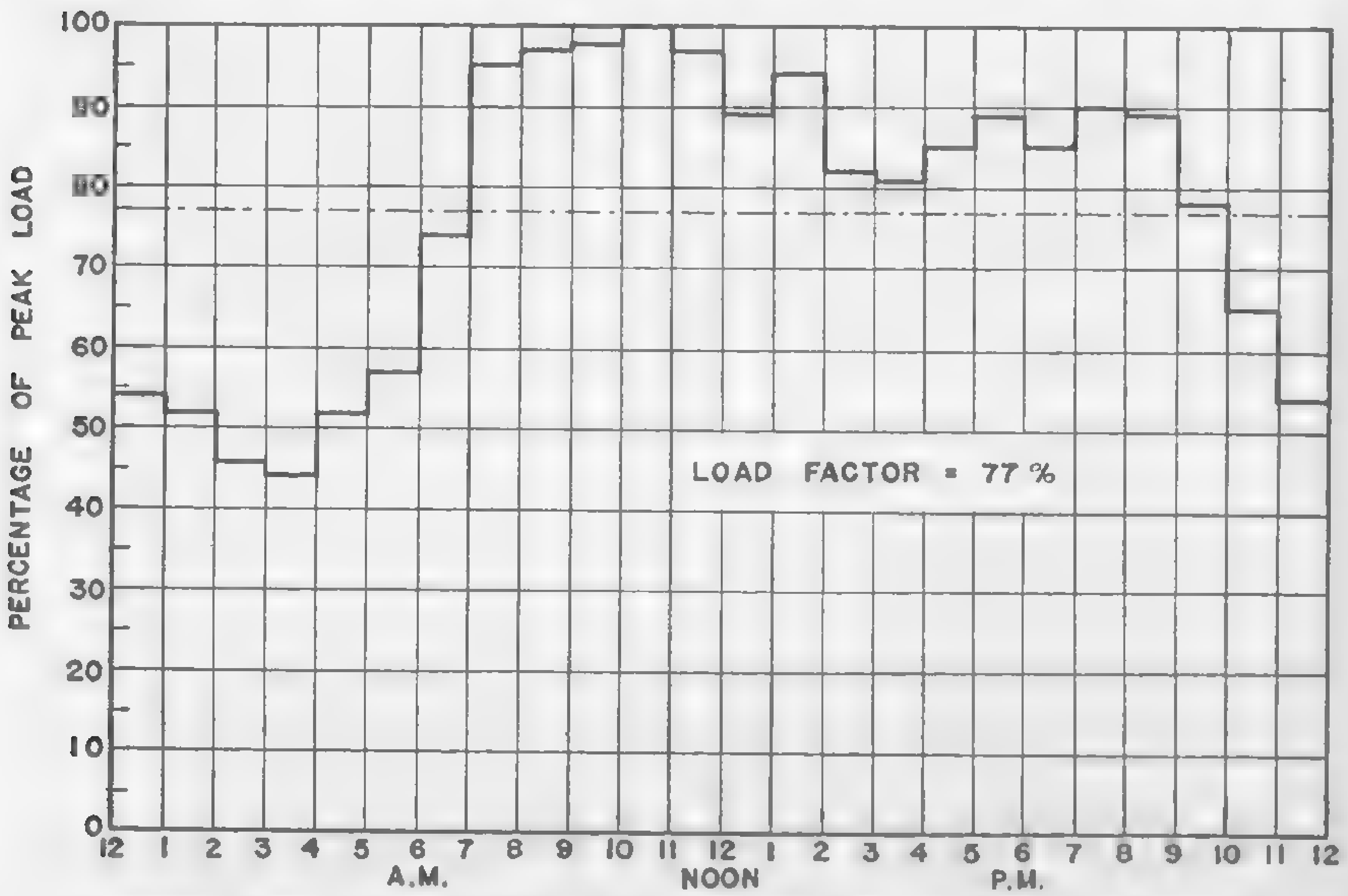
It is assumed that the pondage is provided for meeting the average load demand. For an average discharge of 4020 cfs, the hourly excesses and deficiencies for hourly demands are given in Columns (4) and (5), respectively. The total of these two columns, or 21,306 cfs-hr, should check with each other. This is the cumulative deficiency in 24 hr to be regulated by pondage. Since 1 cfs-hr $= 3600/43{,}560$ or 0.0827 acre-ft, therefore, the pondage required is $21{,}306 \times 0.0827 = 1762$ acre-ft.

When the flow is 5000 cfs, which is greater than the average flow, the hourly excesses and deficiencies are given in Columns (6) and (7), respectively. With the pondage at full capacity at the beginning of draft on the pond at 7:00 A.M., the cumulative pondage at every hour was computed as shown in Column (8). Column (9) gives the hourly spill, which is equal to the discharge in excess of the full pondage capacity. The total spill or losses are equal to 23,320 cfs-hr or $23{,}320 \times 0.0827 = 1930$ acre-ft.

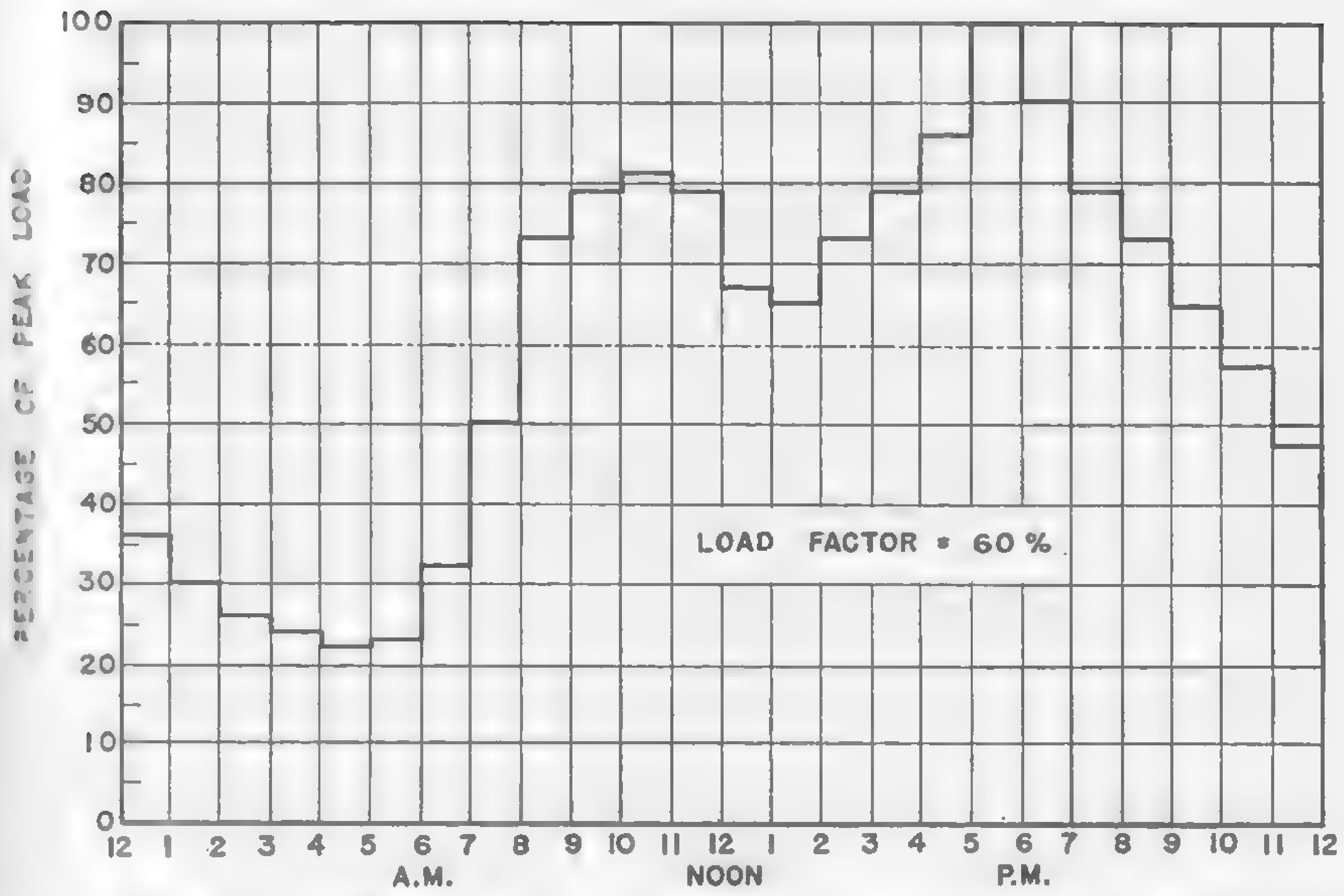
3–10. Storage. The storage required for a given power demand from a known supply is frequently computed by the mass-curve method. The application of this method is explained by the following:



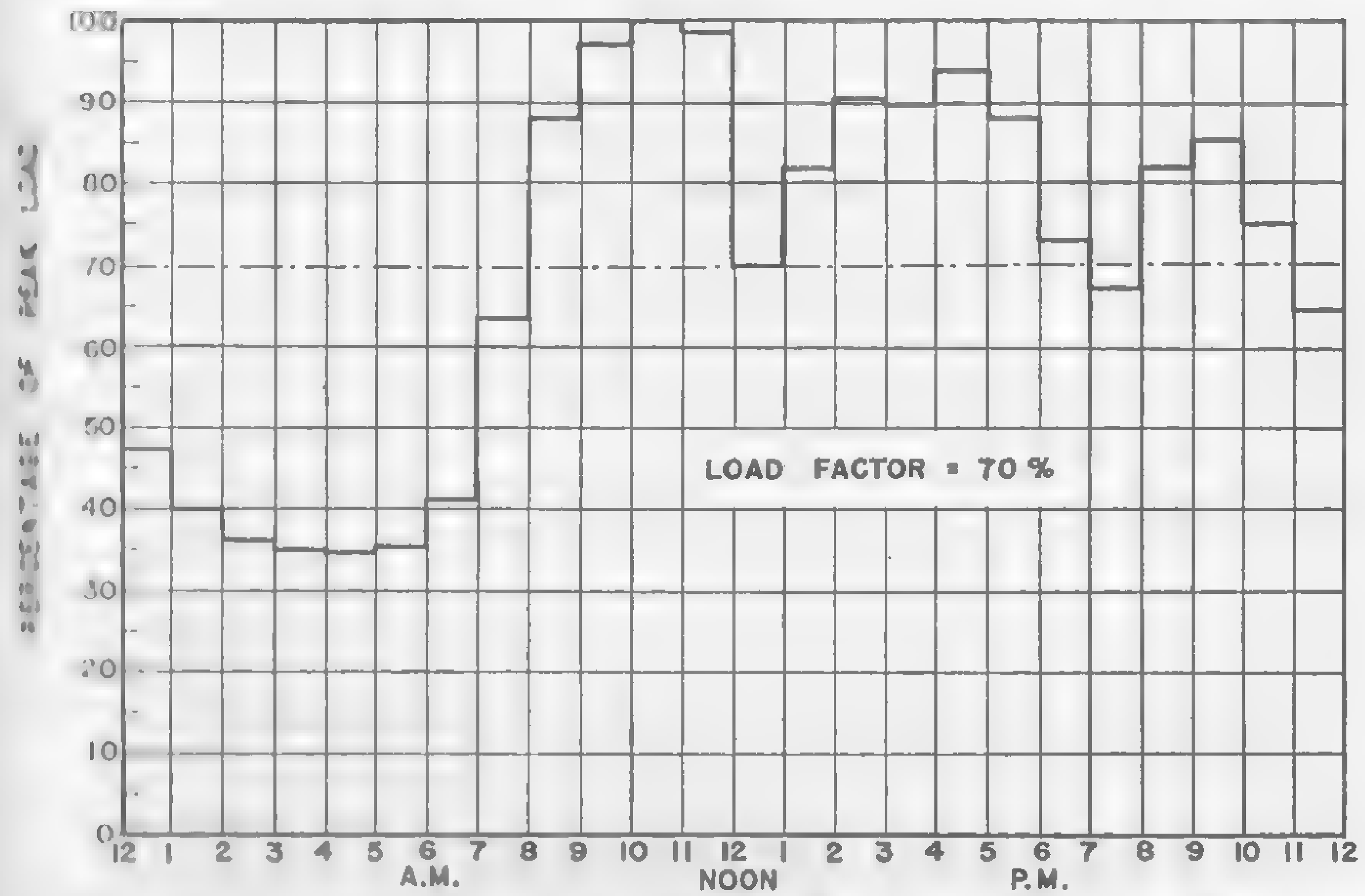
DECEMBER UNIT LOAD CURVE OF A POWER FOR DOMESTIC AND LIGHTING USE



DECEMBER UNIT LOAD CURVE OF A POWER SYSTEM FOR INDUSTRIAL USERS



TYPICAL DECEMBER UNIT LOAD CURVE OF A METROPOLITAN SYSTEM LOAD



TYPICAL AUGUST UNIT LOAD CURVE OF A METROPOLITAN SYSTEM LOAD

FIG. 3-6. Typical peak day unit load curves.

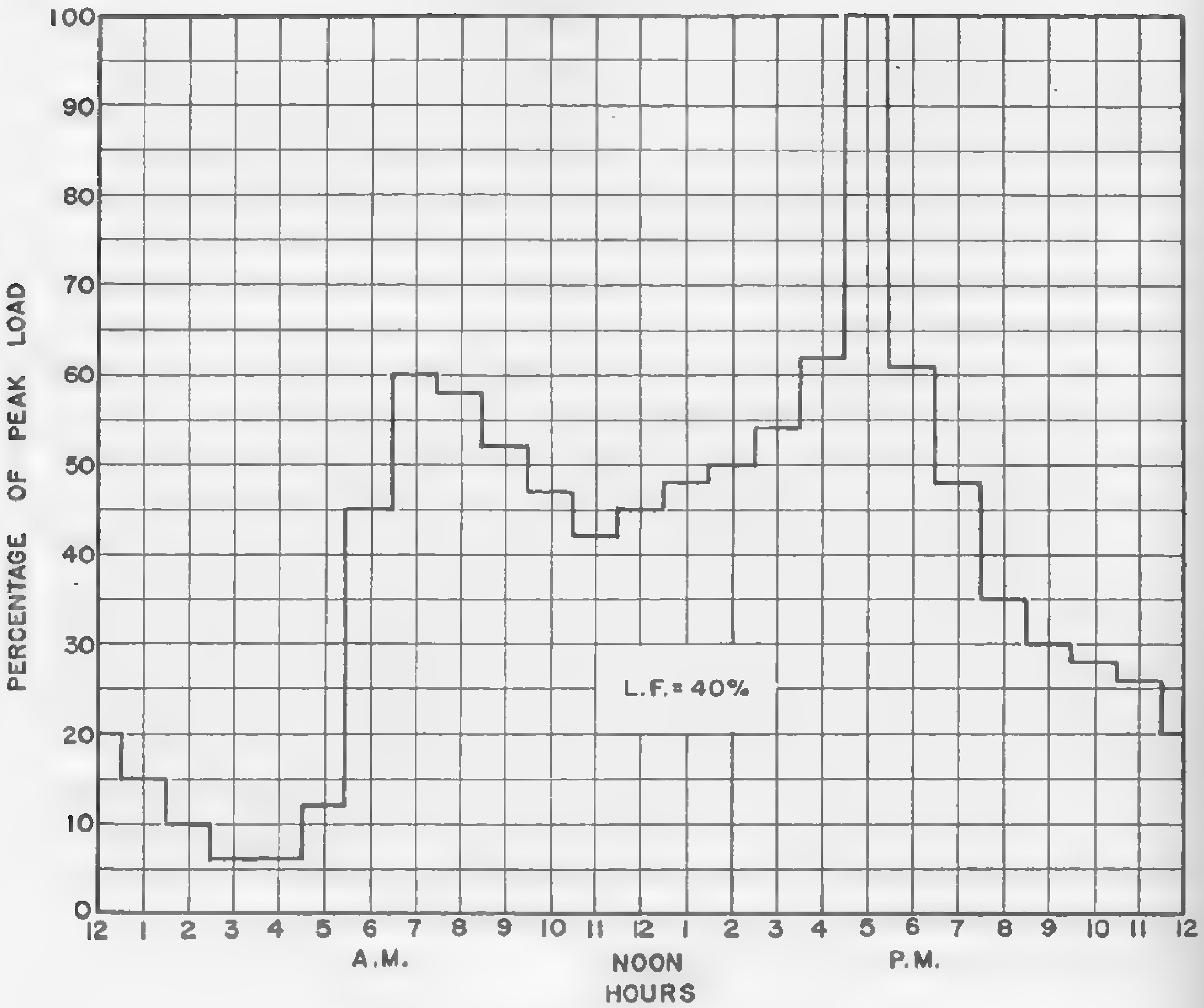


FIG. 3-7. Average daily unit load curve.

Illustrative Example: A mass curve based on the adjusted flow data for the dry period from 1944 to 1947 is plotted as shown in Fig. 3-3. It indicates that during the period from the first week of 1944 to the 52nd week of 1947, 525,651 sec-ft weeks passed the site. The slope of a line connecting any two points on the curve is a measure of the average flow during that period. For instance, the average flow for the dry period from 1944 to 1947 is equal to $525{,}651/4 \times 52 = 2530$ cfs.

From the summit on the mass curve at the 24th week of 1944, it is assumed that the proposed reservoir would be full at that time. The slope of a draft line drawn tangent to the mass curve from the 24th week of 1944 to the summit at the 34th week of 1947 indicates that a draft of 2130 cfs for the dry period of 166 weeks could be sustained if the necessary storage is supplied. The reservoir will be drawn upon at all times when the slope of the mass curve is less than the slope of the 2130-cfs draft line and will be filling when the slope of the mass curve is greater than that of the draft line. The maximum possible draft for the given period is 2130 cfs. The required storage volume to sustain this draft is determined by the height of the maximum intercepted

TABLE 3-4

COMPUTATION OF PONDAGE AND SPILL

Hour	% Peak	Supply = 4,020 cfs			Supply = 5,000 cfs			
		Demand	Excess	Defic.	Excess	Defic.	Capacity	Spill
		Cu ft per sec hr						
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
1 A.M.	15	1,508	2,512		3,492		21,306	3,492
2	10	1,005	3,015		3,995		21,306	3,995
3	6	603	3,417		4,397		21,306	4,397
4	6	603	3,417		4,397		21,306	4,397
5	12	1,206	2,814		3,794		21,306	3,794
6	45	4,522		502	478		21,306	478
7	60	6,030		2,010		−1,030	20,276	
8	58	5,829		1,809		−829	19,447	
9	52	5,226		1,206		−226	19,221	
10	47	4,724		704	276		19,297	
11	42	4,221		201	779		20,076	
12 P.M.	45	4,522		502	478		20,554	
1	48	4,824		804	176		20,734	
2	50	5,025		1,005		−25	20,705	
3	54	5,427		1,407		−427	20,278	
4	62	6,231		2,211		−1,231	19,047	
5	100	10,050		6,030		−5,050	13,997	
6	61	6,131		2,111		−1,131	12,866	
7	48	4,824		804	176		13,042	
8	35	3,517	503		1,483		14,525	
9	30	3,015	1,005		1,985		16,510	
10	28	2,814	1,206		2,186		18,696	
11	26	2,613	1,407		2,387		21,083	
12	20	2,010	2,010		2,990		21,306	2,767
Total	960	96,480	21,306	21,306	33,469	9,949		23,320

ordinate between the mass curve and the draft line. This is found to be 650,000 acre-ft and occurs at the 52nd week of 1946.

The above example pertains to the draft and storage conditions on a total period basis, and assumes that the given flow cycle would be repeated during some future dry period of 166 weeks. It is not always practical to make such long-range assumptions in estimating future available draft and required storage. Study on an annual basis may be more practical and is made in a similar manner. The annual period analysis is based on the driest 52 weeks as indicated by the mass curve. In Fig. 3-3, the driest 52 weeks occurred beginning in the 30th week of 1944 and ending the 30th week of 1945. For this period, the available draft is 1635 cfs and the required storage is 364,000 acre-ft. Another less dry yearly period occurred beginning the 30th week of 1946. This period indicates an available draft of 2510 cfs and a required storage of 476,000 acre-ft. The final determination of the amount of storage depends on good judgment and economical considerations.

CHAPTER 4

HYDRAULIC TURBINES AND THEIR PRELIMINARY SELECTION

4–1. Suitable Range of Head and Specific Speed. Studies of a large number of power installations reveal that there is a range of head and specific speed at which each type of hydraulic turbine is most suitable, and, hence, this range is most commonly adopted by designers. The results of these studies are represented by an experience curve (Fig. 4–1), in which an average relationship between the head and the specific speed for various types of turbines is expressed. The ranges indicated by this curve should facilitate the selection of turbines in a preliminary study of new installations.

It should be noted from the experience curve that over a certain range of head one of three types of runners can be used. The choice requires a careful study of each individual case. Generally speaking, a high-specific-speed runner is more economical because the high speed permits the use of a smaller turbine and generator and, hence, a smaller powerhouse.

The curve indicates a gap in specific speed from $N_s = 8$ to $N_s = 20$. Within this range, runners are rarely designed because of mechanical difficulties in assuring high efficiency. However, in certain cases, such as the design of a turbine to match an already existing generator, the required specific speed may fall within this gap. At the expense of efficiency, this range of specific speed can be secured either by increasing the number of jets for impulse runners, or by designing a very low speed for Francis runners. Of the two ways, the latter is usually found to be more satisfactory. Furthermore, with impulse runners the head range of from 200 to 700 ft is recommended only for small horsepower units, otherwise the generator speed would be too low. On the other hand, in the case of Francis runners, the head range of from 300 to 1000 ft is recommended only for large horsepower units, otherwise the generator speed would be too high.

4–2. Classification of Turbines. Hydraulic turbines may be conveniently classified according to the suitable range of head and specific speed as follows:

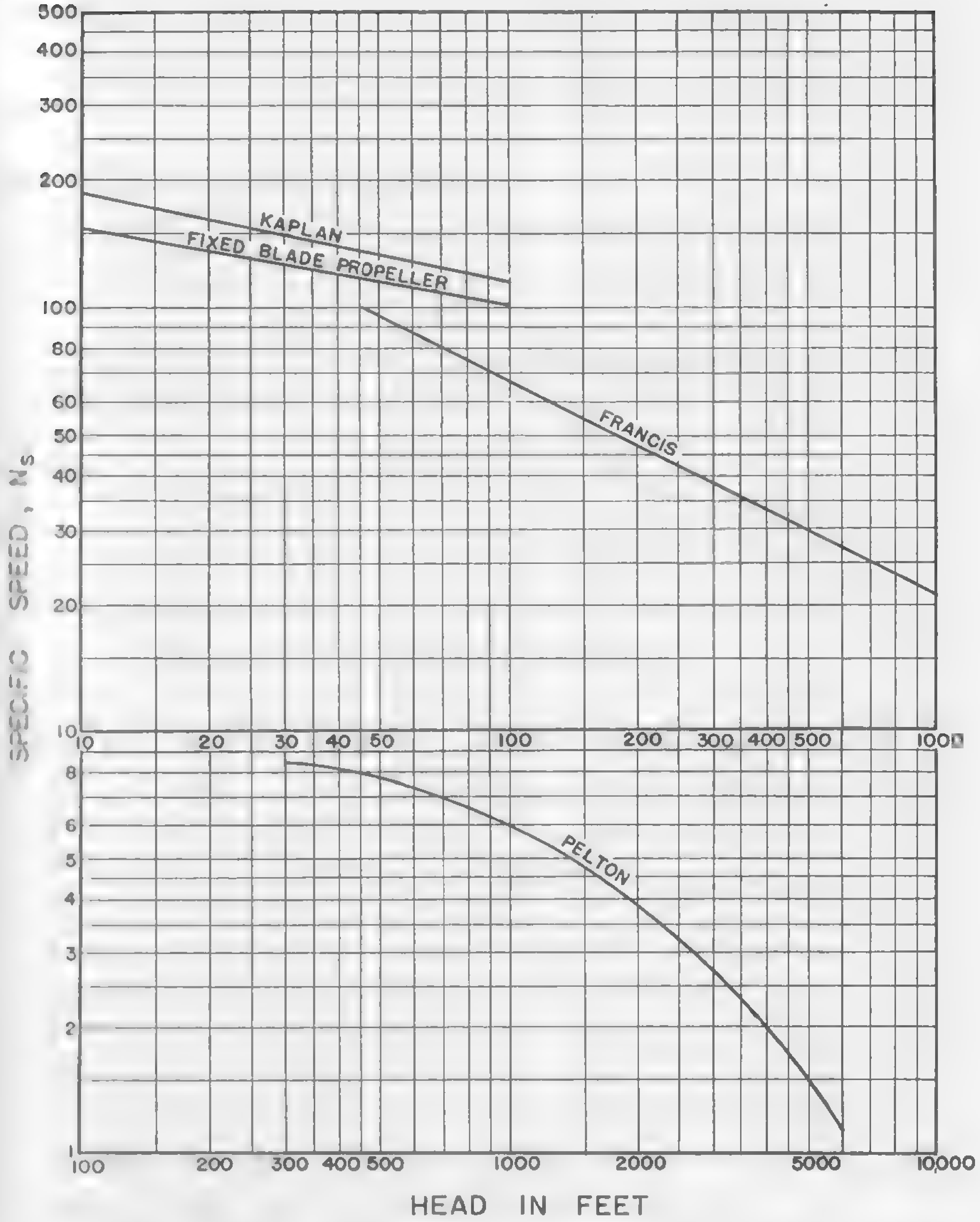


Fig. 4–1. Recommended upper limits of specific speeds for turbines for various rated heads.

1. Classification according to head:

Head, in ft	*Type*
Below 100 ft	Low-head turbine
100 to 1000 ft	Medium-head turbine
1000 ft and up	High-head turbine

2. Classification according to specific speed:

Specific Speed	*Type*
3 to 12	Low-specific-speed turbine
12 to 90	Medium-specific-speed turbine
90 to 250	High-specific-speed turbine

As a rule, low-specific-speed turbines are used for high heads, whereas high-specific-speed turbines are used for low heads.

Low-head turbines are usually of the reaction type, which includes the movable-blade (also called adjustable-blade and Kaplan) propeller, the fixed-blade propeller, and the Francis runners. Propeller-type runners are primarily used for heads below 100 ft because of the relatively deep setting required to minimize cavitation which may shorten the runner life.

For run-of-the-stream operation, the movable-blade-propeller turbine provides valuable peaking capacity by very efficient part-gate operation. The water thus saved is available from storage to be used to meet short-term, high-load demands, during which the generator may be operated at overloads. Normal operation and best efficiency of movable-blade turbines may be at 50 to 75 per cent of full capacity, instead of approximately 80 to 90 per cent for Francis turbines.

Medium-head turbines are usually of the Francis type. However, in installations of small horsepower ratings and where abrasive materials are carried in the water, impulse turbines may be used.

High-head turbines are generally of the impulse type. The impulse type of runner is generally suitable under heads with a range from 800 ft to 3000 or 4000 ft. However, at the Lac Fully power plant in Switzerland, this type of turbine is operating at a head of 5400 ft. More than one jet is used to increase the speed and reduce the physical size and cost of large units.

4–3. Turbine Rating. Hydraulic turbines may be rated under several conditions: maximum head, minimum head, normal head, and design head. The design head is the head at which the runner is designed for best speed and highest efficiency. The design head is fixed, but the performance of the unit at all other heads is determined experimentally. Generally, the design head is selected as the head above and below which the average annual generation of power is approximately

equal. This selection insures the most efficient use of the water when the point of best efficiency is at the average weighted head.

To assure high efficiency of the unit and to avoid excessive cavitation, the maximum and minimum heads of various types of turbines are generally limited in terms of per cent of the design head, as follows:

Type of Turbine	*Minimum Head* %	*Maximum Head* %
Francis turbines	65	125
Fixed-blade-propeller turbines	50	150
Movable-blade-propeller turbines	50	150

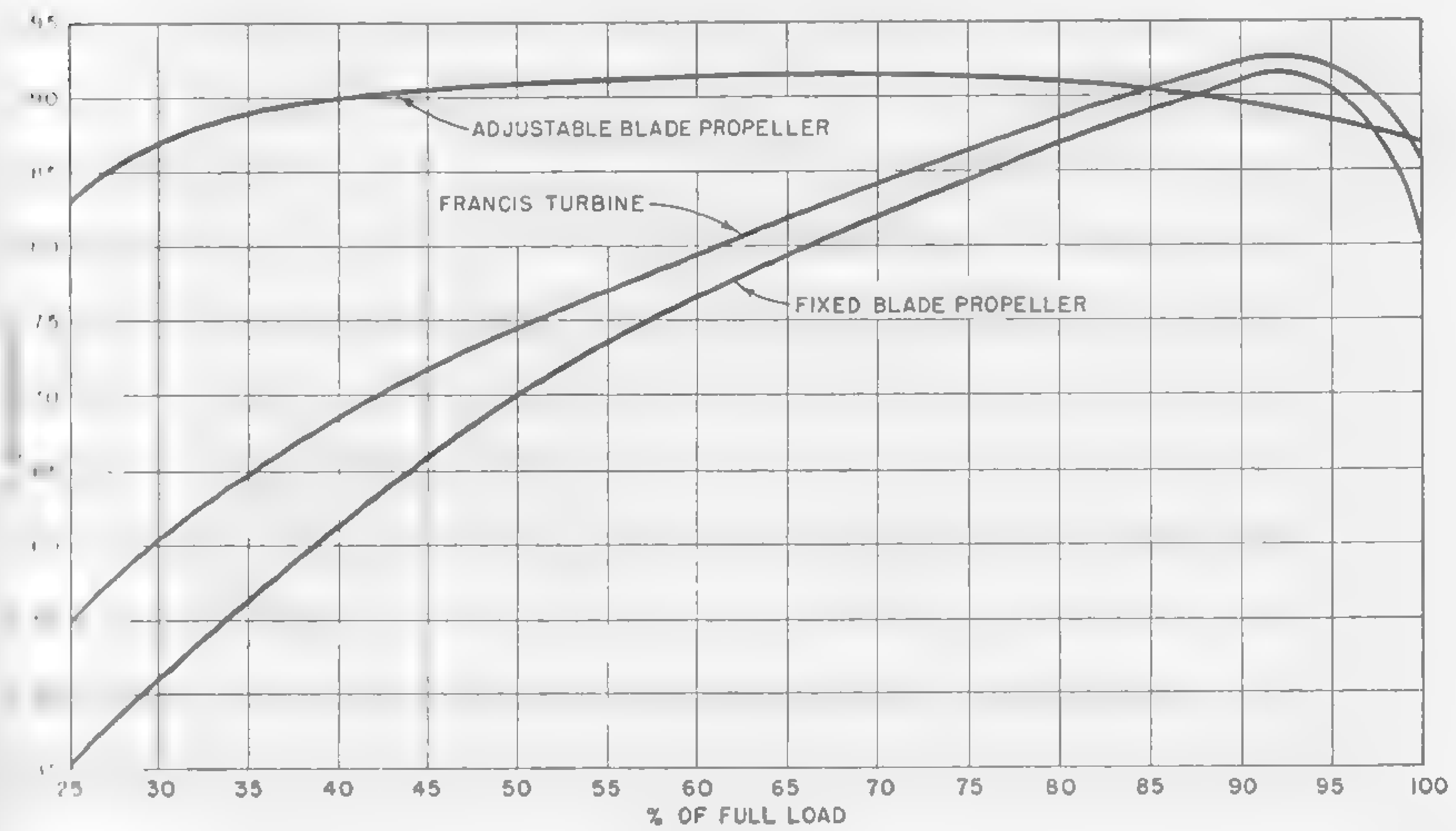


FIG. 4–2. Performance curves for adjustable and fixed-blade propeller and Francis turbines. (S. Morgan Smith Co.)

4–4. Turbine Performance. Because of changes in head and load during operation, it is essential to know the power and efficiency of a turbine at various heads and to evaluate the over-all economics of power developments. Since this information can be determined only from actual tests, the manufacturer usually furnishes such test data in the form of curves similar to those shown in Fig. 4–2 or 4–3. These curves are obtained by plotting the power in percentage of full load against the efficiency. However, the performance curve in a general sense may be any curve showing the relations among such elements as power, speed, discharge, and efficiency of a turbine operating at various conditions. Sometimes it is more convenient to use the turbine constants instead of the actual quantities, particularly for comparing performance data and computing various operating curves. The test required for obtaining a performance curve may be made either upon

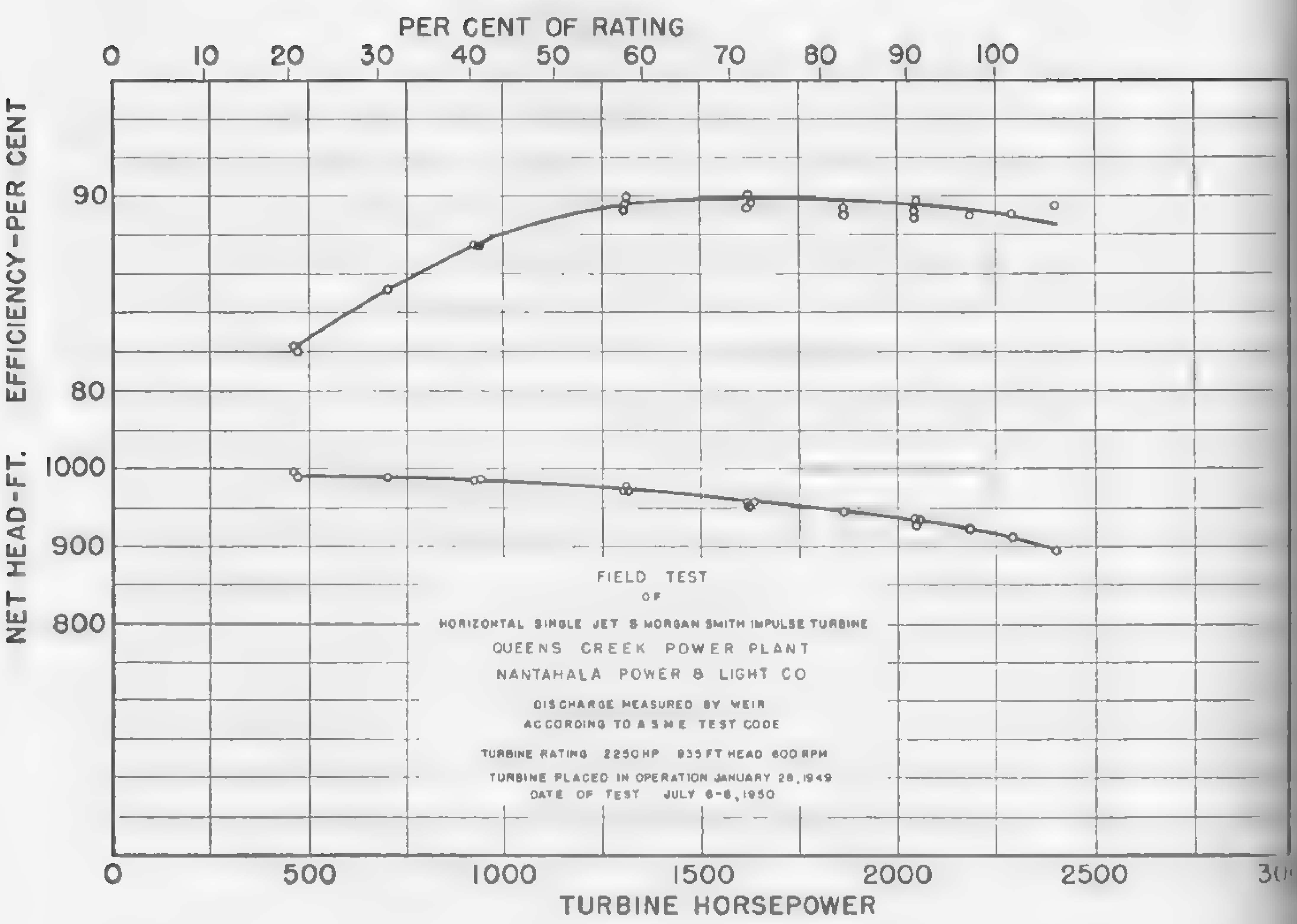


FIG. 4-3. Performance curve for an impulse turbine with $N_s = 5.5$. (S. Morgan Smith Co.)

model or prototype turbines. When the model turbine is used, the performance of the actual turbine can be related to the turbine constants (Art. 2-8) obtained from results of the test. Figure 4-3 shows a typical performance curve of an impulse turbine with a specific speed of

$$N_s = N\frac{\sqrt{P}}{H^{5/4}} = 600\frac{\sqrt{2250}}{935^{5/4}}, \text{ or } 5.5.$$

This curve indicates a maximum efficiency of slightly under 20 per cent at 1600 hp and more than 88 per cent above 1000 hp.

Figure 4-2 shows a comparison of the performance of movable-blade-propeller turbine, fixed-blade-propeller turbine, and Francis turbine, all delivering the same full-gate power under the same head. The movable-blade-propeller runner has a high efficiency in a wide range of load, with an average efficiency that is about 15 per cent higher than either the fixed-blade-propeller or the Francis type.

4-5. Normal Speed. The normal, or operating, speed, N, of a turbine runner may be determined from the preliminary selection of a suitable specific speed from Fig. 4-1. Since $N_s = \frac{N\sqrt{P}}{H^{5/4}}$, then $N = \frac{N_s H^{5/4}}{\sqrt{P}}$. The

value of N thus obtained must be corrected slightly to fit a synchronous speed determined from Eq. (2-26), $N = 120f/P$. When the head on the turbine varies over a considerable range, the lower value of N determined from next higher even number of poles (divisible by 4 for heads up to 600 ft, or by 2 for heads above 600 ft) is used. If the range of heads does not exceed 10 per cent above or below normal head, the next lower number of poles or the higher value of N may be used.

Illustrative Example: It is desired to fix the speed of a turbine which will operate under a design head of 140 ft, maximum head of 155 ft, and minimum head of 115 ft. The power to be developed is 25,000 hp. The frequency is to be 60 cycles per sec.

Solution: From Fig. 4-1 the trial specific speed is 55. The trial rpm $= \frac{55 \times 140^{1.25}}{\sqrt{25,000}} = 167$. The number of poles for a speed of $167 = 7200/167 = 43.1$ poles. Since the variation in head is more than 10 per cent of normal head, 44 poles are chosen. The synchronous speed will then be rpm $= 7200/44 = 163.6$. The specific speed of the design head will then be $\frac{163.6 \times \sqrt{25,000}}{140^{1.25}} = 53.7$.*

A hydraulic turbogenerator installation must operate as closely as possible to a constant speed to maintain control over the frequency of the electric current produced. Changes in the load will cause momentary fluctuations in speed. Changes in gate positions due to changes in load will result in fluctuations in head which will also affect the speed. A governor (see Art. 6-14) is required to control the speed mechanically within a maximum increase or decrease of 30 per cent of normal speed. Speed control involves many factors and requires detailed and laborious calculations to arrive at exact determinations. The factors include the flywheel effect of the rotating machinery, WR^2 (weight times the square of the radius of gyration); the horsepower, P, or KVA of the turbine and generator; the normal speed N in revolutions per minute; the amount of the change in load; the governor time T; the change in head ΔH, and others. The analysis given below gives approximate results to indicate the effect of the WR^2 element; it is not sufficiently refined to give accurate estimates.†

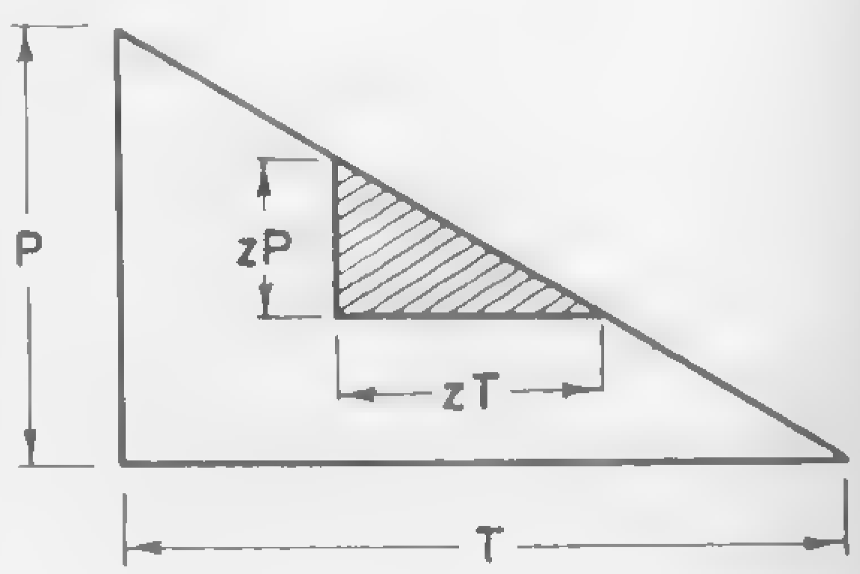
* These data are taken from the S. Morgan Smith turbines installed by the Corps of Engineers at the Dale Hollow Project in Tennessee.

† For precise methods of calculating speed change, the reader is referred to Arnold Pfau, *Hydraulic Turbine Handbook* (3rd ed.; Milwaukee, Wis.: Allis-Chalmers Mfg. Co., 1948); and Earl Strowger and S. Logan Kerr, "Speed Changes of Hydraulic Turbines for Sudden Changes of Loads" (Paper presented at the Spring meeting of the American Society of Mechanical Engineers, San Francisco, Calif., 1926)

The notations used in the following derivation are as follows:

P = the full load horsepower output of the turbine
ΔP = the change in load
z = the part-load factor or the ratio of the change in load to the full load P or $z = \frac{\Delta P}{P}$ or $\Delta P = zP$
T = the total governor time in seconds
W = the weight of the revolving turbogenerator in pounds
R = the radius of gyration of the revolving masses in feet
N = the normal rpm of the turbine
ΔN = the speed change
V = the peripheral speed at the radius of gyration R of the revolving masses in feet per second $= \frac{2\pi RN}{60}$
x = the relative speed change $= \frac{\Delta N}{N}$, which may be positive or negative depending on whether the load is rejecting (speed rise) or increasing (speed drop)
C = the flywheel constant, defined as $C = \frac{WR^2N^2}{P}$

Assume that the change of load is linear with respect to the time, which is very nearly true as shown on the sketch. It indicates that for a partial load change, zP, the corresponding change in time is zT. By the principle of mechanics, the change in kinetic energy due to the partial load change should be equal to the area of the shaded triangle. Expressed in foot-pounds per second, this is equal to



$$\frac{1}{2}(zP)(zT)550 = 275z^2PT$$

(550 fps = 1 hp). This change in kinetic energy must be balanced by the change in kinetic energy of the revolving masses engaged, which is

$$\Delta\left(\frac{WV^2}{2g}\right)$$

Since $V = 2\pi RN/60$, $x = \Delta N/N$ and $C = WR^2N^2/P$, and considering the increment to be very small, the above expression becomes by differentiation

$$\Delta\left(\frac{W\pi^2R^2N^2}{1800g}\right) = \frac{W\pi^2R^2N^2}{900g}\frac{\Delta N}{N}$$

$$= \frac{\pi^2CPx}{900g}$$

Equating the two expressions for kinetic energy,

$$275z^2PT = \frac{\pi^2CPx}{900g}$$

and solving for x,

$$x = \frac{800{,}000z^2T}{C}$$

It is thus apparent that the speed change ratio, x, varies directly as the governor time, T, and the horsepower, P, and inversely as WR^2. The generator furnishes on an average about 95 per cent of the total WR^2 and the turbine about 5 per cent. The average value of WR^2 for generators can be computed roughly by the formula:

$$WR^2 = \frac{700{,}000(KVA)^{1.25}}{(\text{rpm})^2}$$

The speed change due to variations in head can be computed from

$$\text{Speed in per cent of Normal Speed} = \left(1 \pm \frac{\Delta H}{H}\right)^{1/2}100$$

where H = normal head and ΔH = rise or drop in head.

4-6. Runaway Speed. The runaway speed of turbines is the maximum speed that the runner could rotate under a given head and gate opening with no external load (i.e., no torque on the turbogenerator shaft). From this it follows that when the turbogenerator is disconnected from the external load, the energy of water delivered to the turbine is completely converted into kinetic energy by the turbogenerating machinery. In actual operation it is possible that the load on the turbogenerating machinery may be suddenly disconnected. It is essential, therefore, to determine the runaway speed for a generator. The turbogenerating machinery must be designed to withstand the runaway speed. While the turbine runner is usually strong enough to withstand this high speed, the generator is comparatively weak, hence, the assembled rotor must be able to withstand the stresses imposed by the runaway speed of the unit.

The range of the runaway speed, in terms of normal speed, for various types of turbines is generally as follows:

Type of Turbine	*Range* (%)
Kaplan	250 to 300
Francis	200
Impulse	Slightly less than 200

In all types of turbines the runaway speed of a prototype can be predicted from the laboratory test of a homologous runner.

The runaway speed of Kaplan turbines is generally 2.5 to 3 times normal speed, whereas with impulse and Francis turbines it is rarely over twice normal. In both types, the runaway speed of a prototype can be predicted from the laboratory test of a homologous runner by proportioning inversely as the runner diameter and directly as $\sqrt{H}$.

Laboratory tests have shown that the runaway speed is affected by the cavitation characteristic, *plant sigma,* that is,

$$\sigma = \frac{H_b - H_V - H_S}{H} \qquad \text{(See Art. 4–8.)}$$

The effect of the cavitation characteristic on runaway speed is much more pronounced in a Kaplan turbine than in a Francis turbine, as seen by comparing Figs. 4–4 and 4–5. For all practical purposes this effect can be disregarded in a Francis turbine, but with a Kaplan turbine it is very significant.

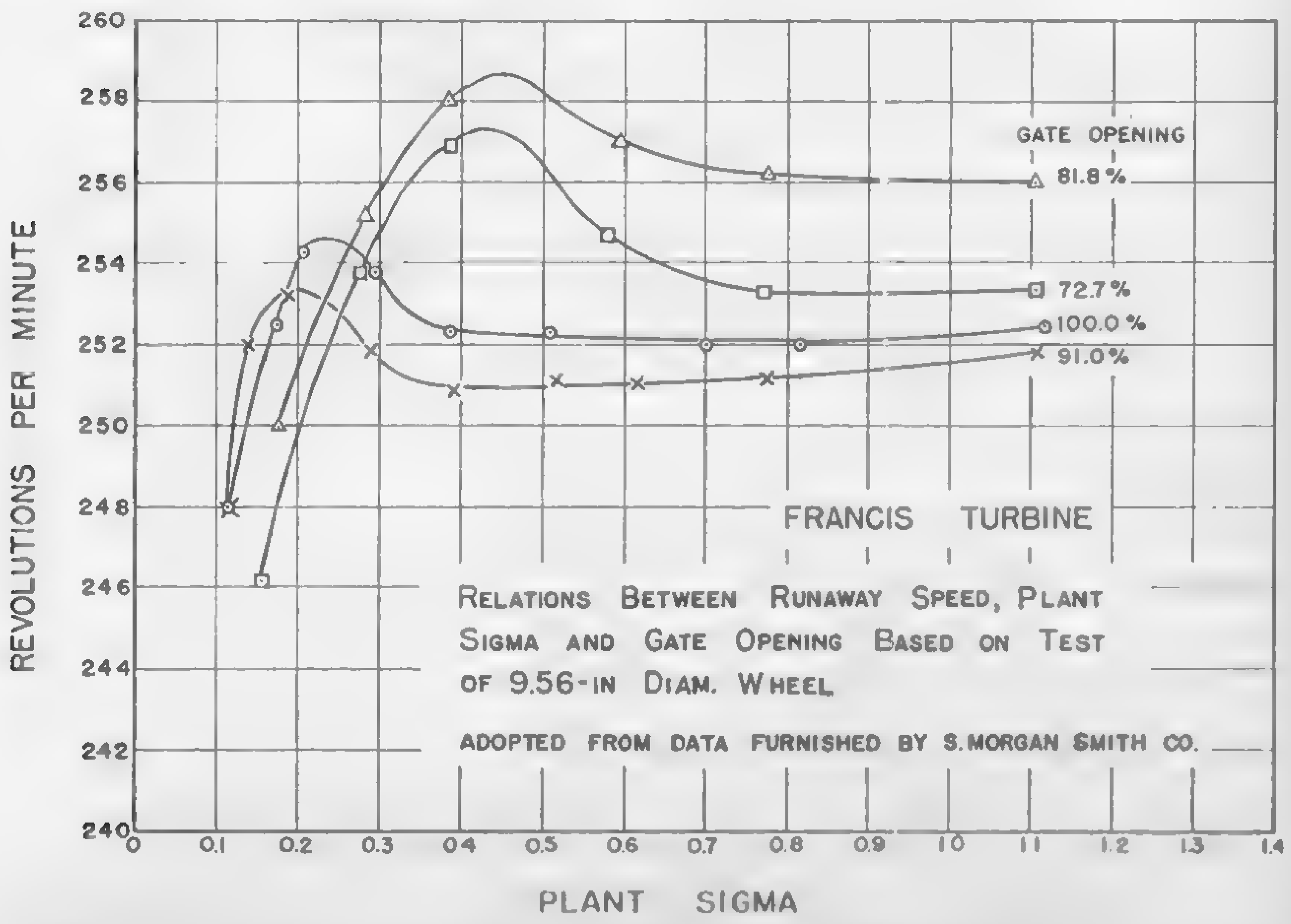


FIG. 4–4. Effect of gate opening on runaway speed, Francis turbine. (Data from S. Morgan Smith Co.)

Runaway speed of both Kaplan and Francis turbines also varies with gate opening. As seen from Figs. 4–4 and 4–5, the maximum runaway speed is not necessarily at full gate opening.

4–7. Cavitation. When the pressure at some point in a stream reaches the vapor pressure, cavities, or vapor pockets may be carried along with the flowing water to the region where the pressure exceeds the vapor pressure, the vapor condenses, and the cavities suddenly collapse. This violent collapse creates hammering pressures concentrated on a very limited area. The pressures can be high enough not only to produce noise and vibration but also to tear the particles of material away from the boundary surface, causing the surface to have a honeycomb appearance (Figs. 4–6A and 4–6B) known as *pitting.* The phenomenon of the formation of vapor pockets is called *cavitation.* As the extent of cavitation increases, the power and efficiency of the unit are greatly impaired.

Laboratory experiments have also shown that cavitation has a certain effect on runaway speed of reaction runners. It is found that the runaway speed becomes greater when the cavitation develops.

Experience has revealed that the pitting of the turbine and its setting occurs most commonly at the following parts:

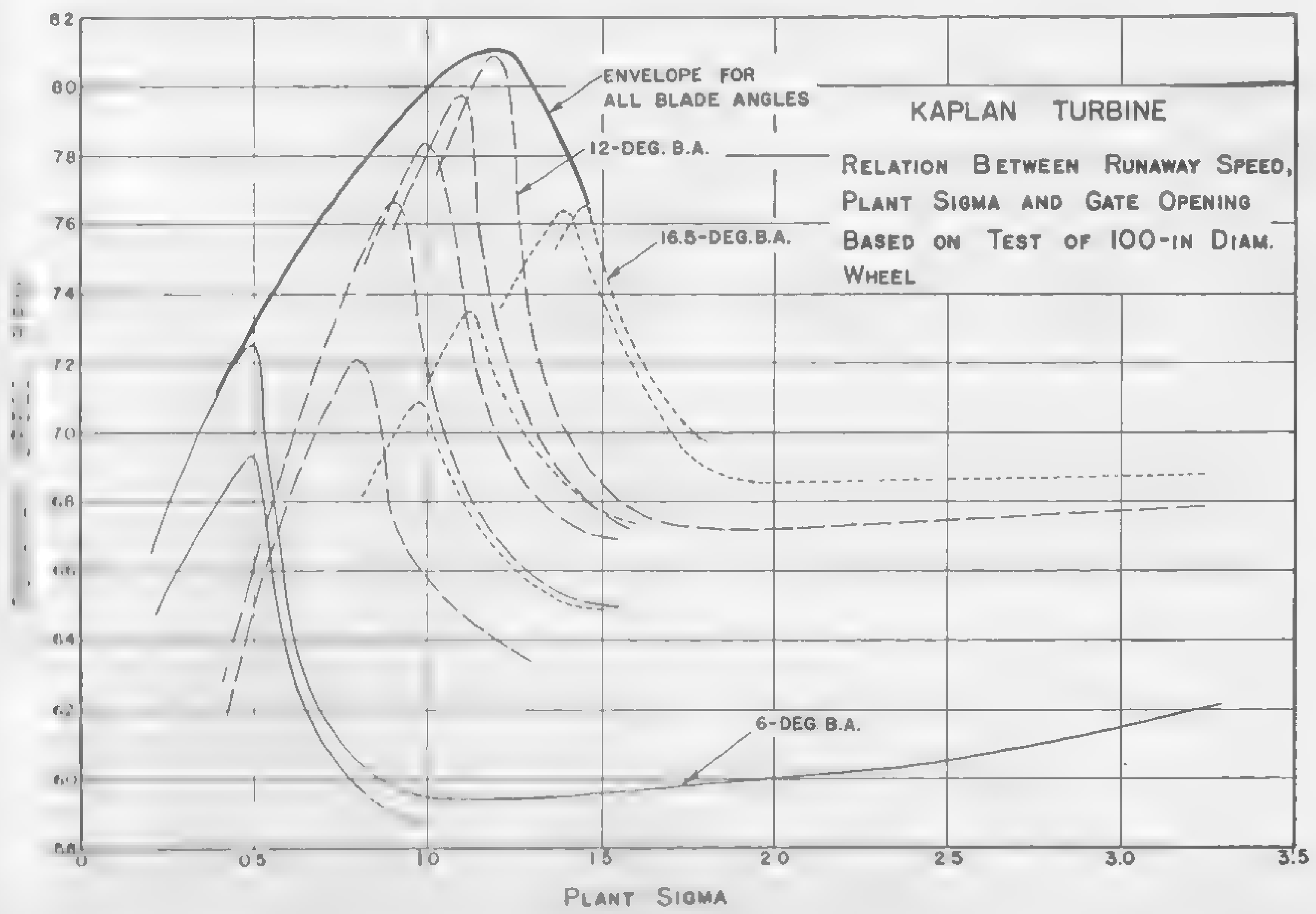


FIG. 4–5. Effect of gate opening on runaway speed, Kaplan turbine. (Data from S. Morgan Smith Co.)

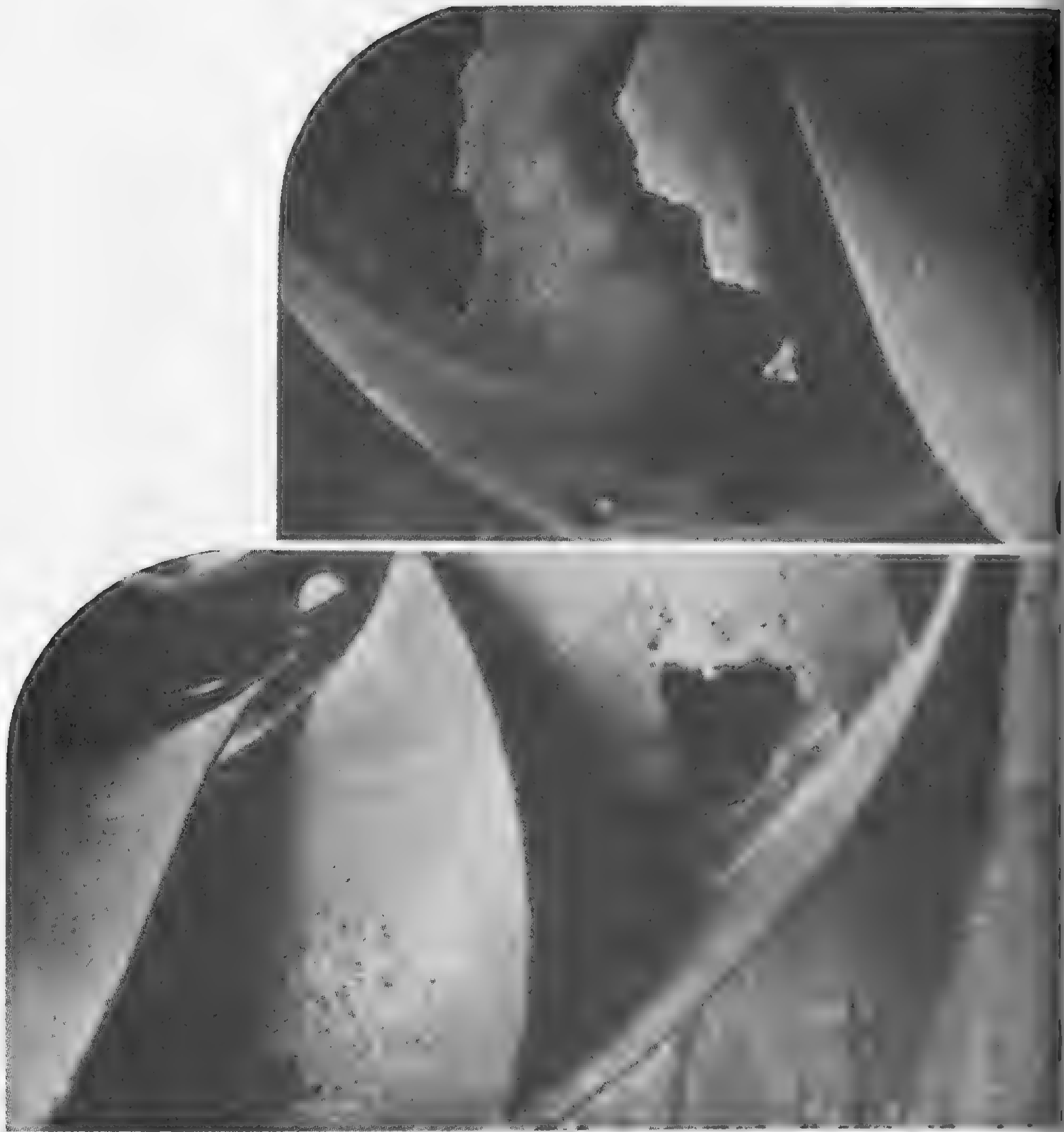


Fig. 4-6A. Effect of cavitation on runner blades. (Baldwin-Lima-Hamilton Corp.)

1. On the back of the bucket or blade of the runner near where the water leaves the runner.
2. On the tip of the needle valves of impulse turbines where operated at or near the critical valve opening of approximately 40 per cent (Fig. 4-6B).
3. On the wall of the upper portion of the draft tube.
4. On any surface where sharp and abrupt corners or any combination of circumstances which will produce uneasy curvature, vortices, eddies, separation, or high local velocities that are conducive to cavitation.

In engineering practice the cavitation may be prevented or reduced by the following methods:

1. To improve the design of the turbine and its setting by providing contours of easy curvature.
2. To employ a tough and resistant material and to use steel liners in conduits in the region where pitting is likely to occur. Experience has shown that a dense and homogeneous material is more resistive to damage than a porous and nonhomogeneous material.
3. To lower the setting of the runner with reference to tailwater so that a low pressure sufficient to cause vaporization will not develop. This will be described further in the following article.

Fig. 4-6B. Effect of cavitation on a needle valve. (Bureau of Reclamation)

4-8. Cavitation Coefficient—Plant Sigma σ. The absolute pressure at a point of minimum pressure in a turbine may be expressed in feet of water, as follows:

$$H_{\min} = H_{\text{atm}} - H_s - \frac{V^2}{2g} = H_{\text{atm}} - H_s - \sigma H \tag{4-1}$$

where H_{atm} = atmospheric pressure, depending on the elevation above the mean sea level at the site when the runner is installed; H_s = draft head, or distance from tailwater to the point of minimum pressure, which point is usually taken as the center line of the propeller runners

or as the bottom of the Francis runners; and $V^2/2g$ = velocity head proportional to the total head H, or $V^2/2g = \sigma H$, in which σ is a coefficient.

As mentioned in the previous article, cavitation will occur when the absolute pressure is equal to vapor pressure, H_V, or

$$H_{atm} - H_s - \sigma H = H_V \tag{4-2}$$

Solving for σ,

$$\sigma = \frac{H_{atm} - H_s - H_V}{H} \tag{4-3}$$

or

$$\sigma = \frac{H_b - H_s}{H} \tag{4-4}$$

in which $H_b = H_{atm} - H_V$ and is equal to the height of the barometric water column. Since H_{atm} depends on the elevation and H_V on the average temperature of the water, this height may be found from the curves shown on the upper right corner of Fig. 4–7 for given values of elevation and temperature.

When a setting of the runner above tailwater is too high or the plant is installed with excessive static draft head, H_s, the cavitation will occur because the absolute pressure by Eq. (4–1) would be low enough to reach the vapor pressure. Equation (4–4) indicates that an excessive H_s which causes cavitation corresponds to a low value of the coefficient σ. It is easily perceived that cavitation occurs when σ is lower than a critical value corresponding to a certain excessive value of H_s. This critical value of σ is known as the *coefficient of cavitation*, or the plant sigma for an existing or proposed plant.

Experience has shown that the plant sigma depends on the specific speed and head of the runner. Figure 4–7 gives the recommended limits of plant sigma for the determination of the safe turbine setting or the distance of the runner above tailwater in preliminary design. These limits are developed from experience with actual installations.

4–9. Position of Runner. The draft head, H_s, is measured from tailwater elevation to the bottom of the runner for Francis turbines, and from tailwater elevation to the center of the runner for propeller-type turbines. The preliminary determination of the elevation of the center line of the distributor for vertical turbines can be easily determined by applying the following procedure:

1. Determine the minimum value of plant sigma from Fig. 4–7.
2. Determine H_b from upper right-hand curve of Fig. 4–7; H_b depends upon average temperature and sea-level elevation.

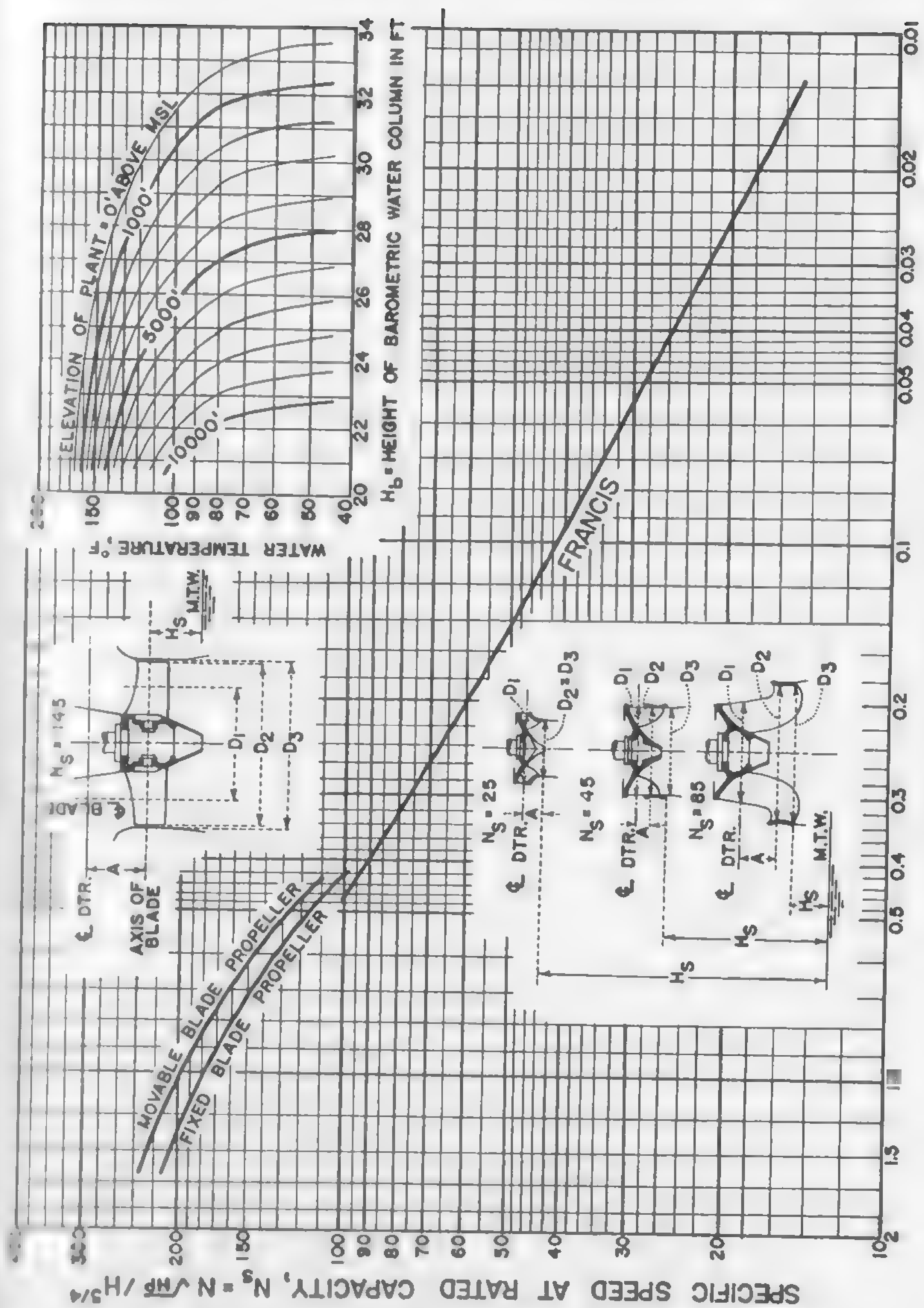


Fig. 4-7. Recommended lowest limits of plant sigma for the determination of turbine setting.

3. Compute H_s. If H_s is negative, the bottom of a Francis runner or the center of a propeller runner must be below tailwater.
4. Determine the approximate distance from the center line of the distributor to the bottom of the runner for a Francis turbine, or to the center line of a propeller runner. Experience curves based on studies of recent plants indicate the following relations:

$$\frac{A}{D_3} = \frac{N_s^{0.34}}{10.7} \quad \text{Francis turbines}$$

$$\frac{A}{D_3} = 0.41 \quad \text{propeller turbines}$$

5. Minimum tailwater elevations depend upon the stage of flow in the tailrace under minimum conditions. If an independent tailrace is provided which carries water only when the plant is operating, minimum stage occurs when only one unit is operating.
6. If $A + H_s$ is added to minimum tailwater elevation, the result will be the elevation of the center line of the distributor.

Illustrative Example: The Hungry Horse Project in Montana, constructed by the Bureau of Reclamation, is expected to operate under a rated head of 400 ft and a maximum head of 479 ft.* The rated horsepower of each unit is 105,000, the normal speed is 180 rpm, the specific speed is 32.4 rpm, the discharge diameter is 126 in., the elevation of minimum tailwater is 3075.5, and the average temperature of the water is 50° F. Determine the approximate elevation of the center line of the distributor.

Solution:

1. From Fig. 4–7 for a specific speed of 32.4, minimum value of plant sigma is 0.063.
2. $H_b = 30$ ft
3. $H_s = 30 - 0.063 \times 400 = 4.8$ ft

$$A = \frac{32.4^{0.34}}{10.7} \times 126 = 38.4 \text{ in.} = 3.2 \text{ ft}$$

4. Minimum tailwater = 3075.5

* The Hungry Horse dam and power plant is located on the South Fork of the Flathead River about 5 miles above the mouth. The dam impounds approximately 3.5 million acre-ft of water with a usable storage of about 3.0 million acre-ft. The hydroelectric power generated (total capacity 285,000 kw) makes possible an increase in the power capacity of the Polson, Thompson Falls, Rock Island, and Bonneville plants by improved river regulation. Construction was started on June 1, 1948 and completed in 1951.

5. 3075.5 + 4.8 + 3.2 = 3083.5 ft above M.S.L. If the maximum head is used the elevation becomes 3078.7

As a comparison of these values with the adopted values, the drawings for this installation show that the center line of the distributor is placed at Elev. 3080 ft M.S.L.; the distance A from the center line of the distributor to bottom of runner is 3.5 ft; and the minimum tailwater elevation with one unit operating is 3075.5 ft. The draft head is therefore 1.0 ft. The actual plant sigma is $\dfrac{30 - 1}{400} = 0.0725$. Since this is greater than the 0.063 minimum allowable, the plant should operate without danger of cavitation.*

4–10. Manufacturers' Information. The following is a list of information needed by a manufacturer for making a proposal to furnish hydraulic turbine machinery.† The parenthetical notes following several of the items indicate the reason for requesting the information.

1. The name, firm or corporation, with address.
2. The location and name of the plant. (Destination assists in determining freight charges.)
3. The approximate elevation of the plant above sea level. (Affects plant sigma, cavitation limitations, setting, runaway speed, and sometimes type of unit.)
4. The total quantity of water in cubic feet or cubic meters per second; with comments regarding the variations in daily and seasonal flow. (Affects the size of unit and if there is great variation in flow, a Kaplan turbine might be recommended instead of a Francis or fixed blade propeller.)
5. The quality of the water. Does it contain sand, chemicals, or other impurities? (A great deal of sand or chemical impurities might cause recommendations for special materials and wearing rings. If the head is moderately high, an impulse turbine might be preferable to a Francis turbine.)
6. The gross head or vertical distance from the headwater level to the tailwater level, with any known variations. (Provides static pressures on wheel case and assists in determining penstock losses.)
7. If it has been determined, give the net or effective head on which all guarantees are to be based, with any known vari-

* *Civil Engineering*, December, 1950, p. 26; and Reclamation Project Data (Washington, D. C.: Government Printing Office, 1948), p. 145.

† This list was adapted from correspondence furnished by the S. Morgan Smith Co., York, Pa., in 1952. Similar information is requested from prospective buyers by all turbine manufacturers.

ations. If it has not been determined, the manufacturer will estimate the net or effective head based on the penstock or flume dimensions.

8. The amount of power desired or required.
9. At what discharge or load is maximum efficiency desired? (Affects size of unit and setting with respect to tailwater elevation.)
10. The number and size of the units contemplated or required now and for future installation. (For more than one unit, there is usually a reduction in average development cost.)
11. The kind and type of machinery to be driven, with data regarding its speed, frequency, and other requirements and whether direct, belt, or geared drive is desired.
12. The direction of rotation required, if already determined—right-hand (clockwise) or left-hand (counterclockwise), when viewed from the driving end. (Frequently, designs for an existing development of the desired size but of opposite rotation are available, and occasionally there is some saving if a new development can be adapted to the general design.)
13. State whether vertical or horizontal setting is preferred or required. (Occasionally, a horizontal setting cannot be used because of cavitation limitations. For small units, particularly if there is a long penstock, a horizontal setting is more economical. Flywheels are not desirable on vertical units and hence, extra WR^2 must be put into the generator rotor. Horizontal units can use a flywheel.)
14. State the distance from low tailwater to powerhouse floor. (Influences the length of turbine shaft for vertical units.)
15. If a supply pipe is required, state the approximate length, diameter, and material, if already designed or installed.
16. If a surge tank is installed or contemplated on the pipe line, give the distance along the penstock from the surge tank to the powerhouse and all available surge tank data. (Items 15 and 16 both affect speed regulation data.)
17. Will the plant operate separately or in parallel with a power system? If in parallel, give approximate installed capacity of the system and its frequency. (Influences the amount of WR^2 to be recommended.)
18. Supplement this information with drawings or sketches, to assist in proper interpretation of the data. (The importance of sketches or drawings giving information on the development cannot be overemphasized.)

CHAPTER 5

DIMENSIONS OF WATER PASSAGES AND POWERHOUSES

5–1. Planning. In any engineering development, the importance of advanced planning cannot be overestimated. The planning of a hydroelectric installation is carried on along with investigations of topography, stream flow, power possibilities and markets, foundation conditions, and project layout. Preliminary planning determines the capacity, number and size of units, and finally may result in preliminary estimates of cost. The economic and engineering feasibility of the project depends largely upon the results of the studies made in connection with the preliminary planning. Turbine manufacturers will insist upon preparing final designs of turbines and water passages after they have been awarded the contract. However, it is unreasonable to expect prospective bidders to spend too much effort and money in the preparation of numerous alternative designs, unless they can be reasonably assured that a contract will ultimately be awarded. It is, therefore, important that most agencies interested in hydroelectric installations have men in their engineering departments who are qualified to arrive at reasonably accurate preliminary project layouts. Such information can then be presented to several manufacturers for consultation and as a basis for making bids on the cost of the machinery.

5–2. Number of Units. Studies made prior to the preliminary design stage will have furnished information on maximum, minimum, and average heads which will prevail at particular times when data are available on storage facilities and high, average, and minimum stream flows. From these studies the total installed capacity may be determined. The next step is to decide how this total installed capacity can be divided among multiple turbine units if more than one unit is indicated. It is not practicable to formulate an easy rule for selecting the number of units or the horsepower rating of each. A study of the expected load growth may be helpful. The advantages of different-size turbines to fit load demands are usually offset by the disadvantages of high cost and the noninterchangeability of spare parts. A study of the load curve should include a consideration of the length of time that the turbine or turbines would run at part load. Some types of runners produce power at relatively low efficiencies at part load (see Fig. 4–2).

The accessibility of the site may also determine the size of the units because of limitations imposed by transportation and handling facilities. It may be necessary to consult turbine manufacturers before a final decision is reached.

In an isolated plant or one which is not interconnected with any other power supply, it may be necessary to provide reserve capacity for carrying the load during temporary partial shutdowns. In interconnected plants involving steam association, reserve capacity is usually supplied by steam units. Experience and judgment are important to the proper selection of units. An inquiring student can acquire these qualities by studying existing installations.

5–3. Generator Capacity. The generator capacity is dependent upon the capacity of the turbines. Best possible efficiency during low-flow periods is obtained when the generator capacity is approximately equal to the output of the turbine at best efficiency under average head conditions. It is sometimes advisable to provide reserve capacity in the generator. Running the turbines at full gate with a temporary loss of efficiency would provide additional power to offset the loss of one unit undergoing repairs.

5–4. Type of Runner. The selection of type of turbine runner usually depends upon the head. The Pelton type is generally selected for heads above 800 to 900 ft. Francis-type runners are suitable for heads of 70 to 900 ft. High-speed propeller-type runners are usually used for heads less than 70 ft. These values overlap somewhat and should not be considered as rigid. Pelton wheels have been installed to operate under heads as low as 200 ft, and propeller-type runners have been selected for heads slightly more than 100 ft. The upper limit of 70 ft for propeller runners is too high for small runners. In general, propeller-type runners of less than 10,000 hp should not be used for heads greater than 30 ft. At the McNary Project, Oregon and Washington, the Corps of Engineers installed in 1953–54 fourteen Kaplan turbines, each having a rated head of 80 ft, maximum head of 92 ft, and rated horsepower of 111,300.

The specific speed, N_s, is also a factor in the consideration of the type of runner. The range of specific speeds for the Pelton type (single jet) is from 1.5 for a 5000-ft head to 9 for a 100-ft head. Specific speeds for Francis-type runners range from about 15 for a 1000-ft head to 100 for a 50-ft head. For propeller types the range is from 100 at a 75-ft head to 200 at a 20-ft head. It must be emphasized that these figures are only representative values and they should not be interpreted rigidly.

5–5. Determination of Runner Diameter. The discharge diameter of the runner of vertical Francis and propeller-type runners provides a very useful base figure which can be used to determine preliminary key dimensions of the scroll case and draft tube and also the over-all dimensions of the substructure of a powerhouse. Reasonably accurate runner discharge diameters, D_3, can be determined in the following manner:

1. The effective head, H; rated horsepower, hp; speed, rpm; specific speed, N_s; and runner type are selected.

2. The model ratio, m, of the given turbine to a homologous turbine which will deliver 1 hp under 1-ft head may be calculated by the formula

$$m = \frac{\sqrt{\text{hp}}}{H^{3/4}}$$

3. The specific diameter, D_s, in inches or the diameter of the homologous turbine (delivering 1 hp under a 1-ft head) may be calculated by the following experience formulas:

$$D_s = \frac{129}{N_s^{0.37}} \quad \text{Francis runners}$$

$$D_s = \frac{113}{N_s^{0.34}} \quad \text{propeller runners}$$

4. The required runner discharge diameter in inches, D_3, may then be computed by the expression

$$D_3 = mD_s$$

Example: Determine the runner discharge diameter of a Francis runner which will deliver a rated horsepower of 105,000 under a head of 400 ft and operate at a speed of 180 rpm.

$$N_s = \frac{180\sqrt{105{,}000}}{400^{5/4}} = 32.6$$

$$D_s = \frac{129}{32.6^{0.37}} = 35.6$$

$$m = \frac{\sqrt{105{,}000}}{400^{0.75}} = 3.62$$

$$D_3 = 35.6 \times 3.62 = 129 \text{ in.} = 10 \text{ ft } 9 \text{ in.}$$

The diameter of an impulse or Pelton runner is not significant in the determination of powerhouse dimensions. However, the pitch diameter

TABLE 5–1

DATA ON FEDERAL HYDRO PLANTS

Number	Plant	River	State	Capacity HP	Capacity KVA	No. Units	RPM	Rated Head, ft.	Specific N_s Speed	Discharge Diameter D_3, ft.	Area at D_3, sq. ft.	Outside Diameter of Generator, ft.	Spacing of Units, ft.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
				Tennessee Valley Authority (Francis)									
1	Apalachia	Hiwassee	Tenn.	53,000	40,000	2	225	360	33	8.53	59.7	30	44
2	Cherokee	Holston	Tenn.	41,500	33,333	4	94.7	100	61	14.83	173	38.3	61
3	Fontana	Little Tenn.	Tenn.	91,500	75,000	3	150	330	32	12.08	114.7	36	56
4	Norris	Clinch	Tenn.	66,000	56,000	2	112.5	165	49	13.79	149.2	41.7	60
5	Ocoee No. 3	Ocoee	Tenn.	33,500	30,000	1	200	280	32	8.09	51.5	29	—
				Bureau of Reclamation (Francis)									
6	Hoover	Colorado	Ariz.-Nev.	115,000	82,500	15	180	487	27.1	11	95	40	40
7	Hoover	Colorado	Ariz.-Nev.	55,000	40,000	2	257.1	465	27.3	7.25	41.3	28	—
8	Grand Coulee	Columbia	Wash.	150,000	108,000	18	120	325	33.6	14.33	161.4	45	65
9	Parker	Colorado	Ariz.-Cal.	40,000	30,000	4	94.7	77	82.5	15.56	190	40	66
10	Shasta	Sacramento	Cal.	103,000	75,000	5	138.5	330	32	13	132.6	40	57
11	Keswick	Sacramento	Cal.	34,600	25,000	3	94.7	75	78	15.42	186.8	40.2	65
12	Davis	Colorado	Ariz.-Nev.	62,200	45,000	5	94.7	120	59	16.28	208	44	72
13	Hungry Horse	Flathead	Montana	105,000	75,000	4	180	400	33.4	10.50	87.2	38.5	60
14	Seminoe	No. Platte	Wyoming	15,000	12,000	3	225	168	45.6	7.13	40	16.2	30
15	Elephant Butte	Rio Grande	N. M.	11,500	9,000	3	257	132	62.5	6.11	29.3	21	30
16	Green Mtn.	Blue	Colo.	15,000	12,000	2	257	185	45.8	6.29	31.1	21	31
17	Anderson Ranch	Boise	Ida.	18,500	15,000	3	277	245	39	6.29	31.1	20.5	33
18	Heart Mtn.	Shoshone	Wyo.	8,300	8,300	1	450	265	40	3.85	11.6	13.5	—
19	Kortes	No. Platte	Wyo.	18,500	13,333	3	240	200	43.5	6.40	32.2	16.1	32
20	[illegible]	Big Thompson	Colo.	21,000	16,667	3	400	482	25.8	4.82	18.2	19.5	35
21	Marys Lake	Big Thompson	Colo.	11,300	9,000	1	327	205	44.5	5.12	20.6	20	—
22	Canyon Ferry	Missouri	Mont.	23,500	16,667	3	150	125	55.5	9.56	70	—	46
23	Boysen	Big Horn	Wyo.	10,500	8,333	2	180	96	60.5	7.58	45.1	—	40
				Corps of Engineers (Francis)									
24	Alatoona	Etowah	Ga.	50,000	40,000	2	112.5	135	54.5	13.76	148.1	38	54
25	Clark Hill	Savannah	Ga.-S. C.	55,000	44,444	7	100	136	50.5	14.83	173	41.4	62
26	Detroit	N. Santiam	Ore.	70,000	55,555	2	163.6	285	37.3	10.83	92	36.7	54
27	Chief Joseph	Columbia	Wash.	100,000	67,368	10	100	165	53.5	15.50	188.7	42.5	70
28	Denison	Red	Okla.-Tex.	56,000	36,842	2	90	102	65.5	16	201	38.3	64
29	Norfolk	N. Fork	Ark.	42,000	38,889	2	128.6	160	46.2	12.17	116	36.3	54
30	Bull Shoals	White	Ark.	52,000	42,100	4	128.6	190	41.2	12.75	127.8	40	54
31	Whitney	Brazos	Tex.	20,700	16,667	2	128.6	92	65.3	10.92	93.3	34	46
32	Tenkiller	Illinois	Okla.	23,500	17,895	2	150	132	51.6	9.67	73.5	22.25	43
33	Fort Gibson	Grand	Okla.	16,000	12,500	4	100	59	77.1	12.58	124.2	32.8	53
34	Dale Hollow	Obey	Tenn.	25,000	20,000	2	163.6	140	53.4	9.50	71	28.3	42
35	Center Hill	Caney Fork	Tenn.	62,500	50,000	3	105.9	160	46.5	14.58	166.8	40	58
36	Wolf Creek	Cumberland	Ky.	62,500	50,000	6	105.9	160	46.5	14.58	166.8	40	58
37	John Kerr	Roanoke	Va.	45,000	35,555	6	85.7	90	80	16.75	220.5	42.5	70
38	John Kerr	Roanoke	Va.	17,000	13,333	1	138.5	90	80	10.08	79.8	31.7	—
39	Philpott	Smith	Va.	9,400	7,500	2	277	152	61.7	5.42	23.1	14.6	32
40	Narrows	Little Mo.	Ark.	12,000	9,444	2	225	132	59.3	6.75	35.8	—	32
41	Blakely Mtn.	Ouachita	Ark.	52,000	41,667	2	120	168	45.2	13	132.4	40	56
42	Fort Peck	Missouri	Mont.	50,000	38,889	2	128.5	170	46	12.17	116	36	46
43	Fort Peck	Missouri	Mont.	20,000	16,667	1	163.6	140	48	8.33	59.1	25.5	—
44	Fort Randall	Missouri	S. D.	57,500	42,106	8	85.7	112	57.2	15.50	188.7	43.3	70
				TVA (Propeller)									
45	Wheeler	Tenn.	Ala.	45,000	36,000	8	85.7	48	144	22	380	43	76
46	Pickwick	Tenn.	Tenn.	48,000	40,000	6	81.8	43	163	24.36	465	44	80
47	Guntersville	Tenn.	Ala.	34,000	27,000	4	69.2	36	145	22.10	384	46.4	78
48	Chickamauga	Tenn.	Tenn.	36,000	30,000	4	75	36	161	22	380	46.25	80
49	Watts Bar	Tenn.	Tenn.	42,000	33,333	5	94.7	52	139	19.50	298.6	39.5	73
50	Fort Loudoun	Tenn.	Tenn.	44,000	35,555	4	105.8	65	120	18.50	268.8	39.75	70
51	Kentucky	Tenn.	Ky.	44,000	35,555	5	78.3	48	130	21.68	368.5	44	77.5
				Bureau of Reclamation (Propeller)									
52	Drop 3	All Am. Canal	Ariz.	7,500	6,000	2	100	26	—	14.19	156	23.3	52
53	Drop 4	All Am. Canal	Ariz.	13,300	12,000	2	150	49	—	11.83	110	24	43
54	Minidoka	Snake	Idaho	7,000	6,000	1	180	44	—	9.17	66	16	—
				Corps of Engineers (Propeller)									
55	Jim Woodruff	Apalachicola	Fla.-Ga.-Ala.	14,000	11,111	3	75	27	147.5	17.68	245	—	65
56	Bonneville	Columbia	Ore.-Wash.	66,000	48,000	2	75	50	144	23.35	427.8	48	82
57	Bonneville	Columbia	Ore.-Wash.	74,000	60,000	8	75	60	122	23.35	427.8	48	82
58	McNary	Columbia	Ore.-Wash.	111,300	73,684	14	85.7	80	119	23.35	427.8	53	86

or diameter of a circle passing through the centers of the buckets may be determined from the expression:

$$D_1 = \frac{830\sqrt{H}}{\text{rpm}} \text{ in.}$$

The value of ϕ or ratio between the peripheral speed of the buckets to the theoretical spouting velocity of the water under the actual head is usually about 46 per cent. The velocity coefficient of the jet is about 98 per cent. The linear velocity, V_w, of the wheel at the center line of the buckets would then be:

$$V_w = 0.98 \times 0.46 \times \sqrt{2gh} = 3.61\sqrt{H} \text{ fps}$$

or

$$V_w = 216.6\sqrt{H} \text{ fpm}$$

since

$$\frac{\pi D_1}{12} \times \text{rpm} = 216.6\sqrt{H}$$

then

$$D_1 = \frac{830\sqrt{H}}{\text{rpm}} \text{ in.} \quad \text{(approx.)}$$

Table 5-1 gives information on various plants designed by federal agencies.

5-6. The Scroll Case. The scroll case is the most commonly used form of casing for reaction turbines of modern installations. The case is generally built of reinforced concrete or metal. The metals used for this purpose are plate steel, cast steel, and cast iron. The selection of the material of construction depends on the head and capacity of the turbine. Figure 5-1 indicates the approximate ranges in head and horsepower for which the various types of casing are ordinarily used.

The water velocity and the smoothness of flow in the case and at the outlet of the turbine are the most important factors in turbine design, for they affect the efficiency and steadiness of the operation of the unit. The allowable water velocity depends on the head and the material of which the case is made. Figure 5-2 shows the allowable full-load velocities in casings for the various types of units, based upon experience. The horizontal portion of the curves indicates the maximum allowable velocity. However, it is desirable that the velocity should not exceed a certain percentage of the spouting velocity at full head. The recommended value of this percentage for concrete scrolls is about 15 per cent, and for metal scrolls, about 20 per cent.

For smoothness of flow, it is essential that the boundary surfaces of the turbine case be as smooth and continuous as possible. Also, the line

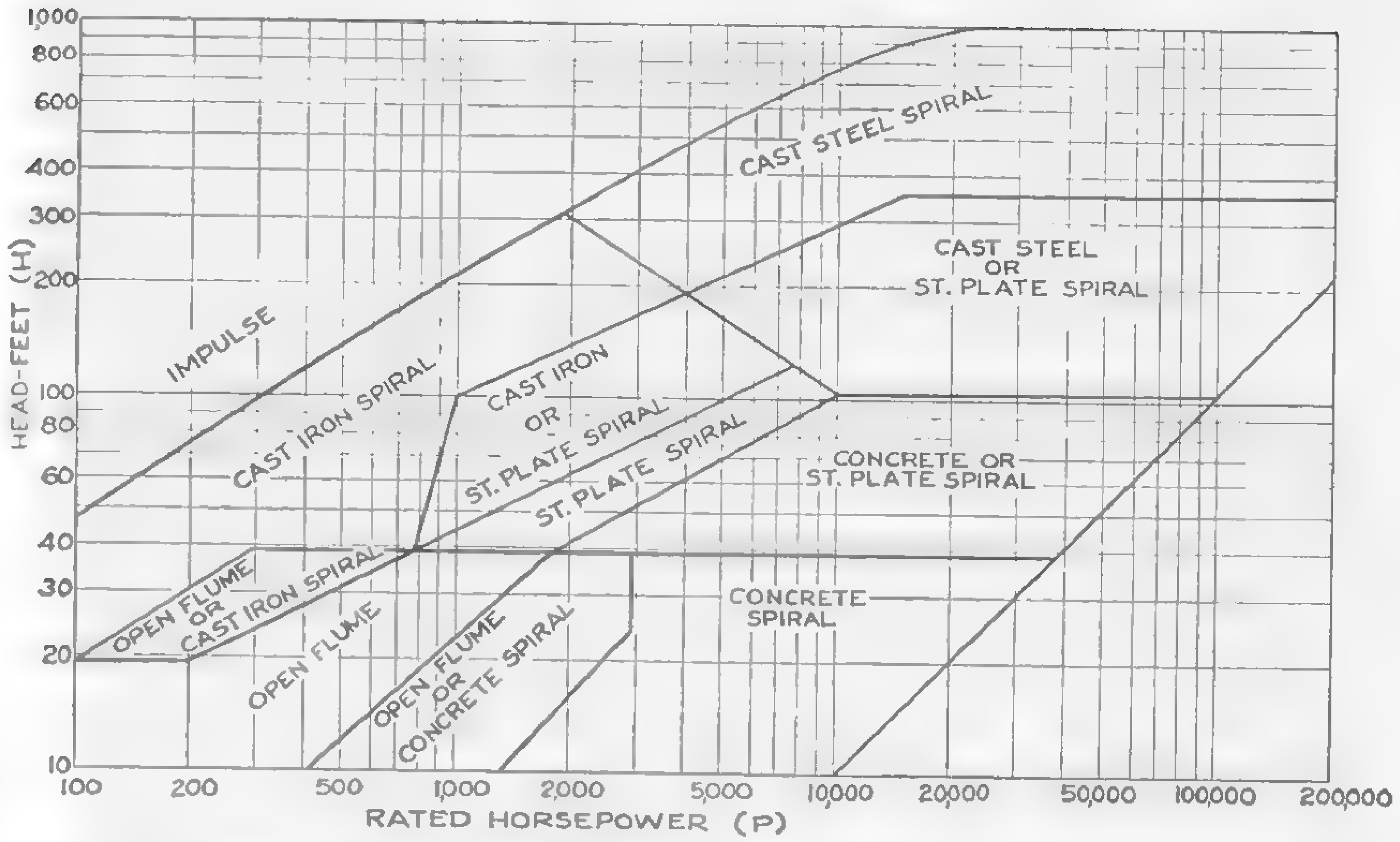


FIG. 5-1. Range in power and head for various types of turbine settings. (Newport News Ship and Dry Dock Co.)

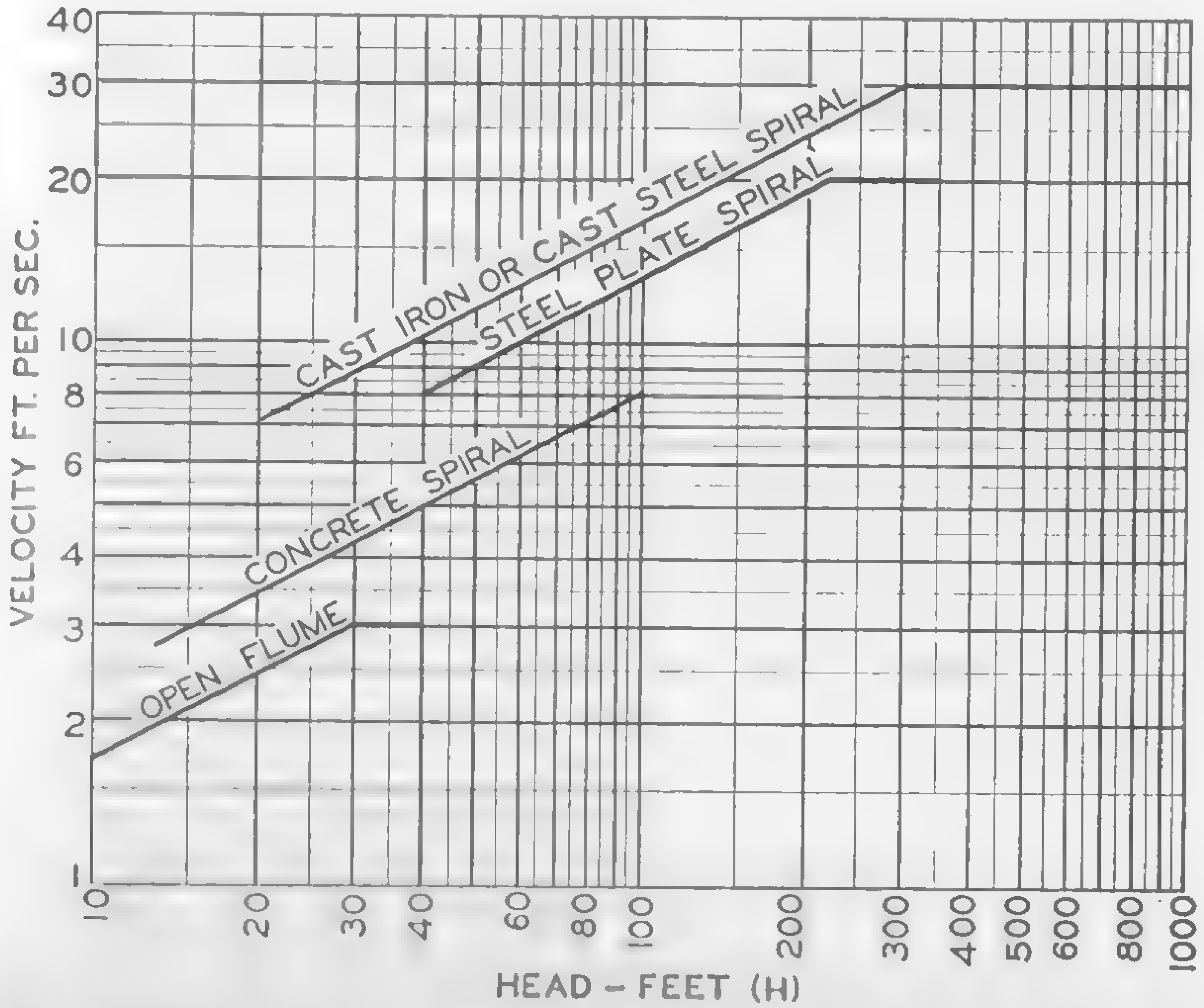


FIG. 5-2. Velocity curves for casings. (Newport News Ship and Dry Dock Co.)

connecting the centers of areas of all sections of the case must form a smooth curve.

A full scroll is generally recommended for high-head installations of moderate capacity, which is commonly the case for Francis turbines with metal scroll. For low-head installations of fairly large capacity, a semiscroll is preferable because it provides sufficiently low velocity with less increase in unit spacing. This is usually the case for propeller turbines with concrete scroll.

Scroll-case dimensions for Francis-type turbines are shown in Fig. 5–3A, and for propeller-type runners, in Fig. 5–3B. Figures 5–3A and 5–3B show values of acceptable ratios between the given dimensions and the outlet diameter of the runner. The ratios shown are based upon studies of a large number of modern plants, a few of which are given in Table 5–1. It will be noted that the given ratios do not apply exactly in all cases, but the percentage of error is rather small.

The theoretical proportioning of the sections of the case may be based on one of the following three methods:

1. Assume a constant angular velocity about the center of the turbine. (Method of constant angular velocity.)
2. Proportion the sections evenly between those determined by the method of constant angular velocity and those by the law of free vortices, i.e., the product of the velocity and the radius is a constant. (Method of modified free vortex.)
3. Proportion the sections such that the pressure around the entire periphery of the turbine runner is approximately equal. This is achieved by reducing the velocity in the scroll case in the direction of flow by an amount sufficient to compensate for the loss in head due to friction. (Method of equal pressure.)

The first two methods usually give the most satisfactory results.

The cross section of the metal scroll case is generally circular or elliptical. For low heads, a rectangular cross section with fillets or rounded corners, Fig. 5–4, is recommended because it will provide sufficiently low velocities. For large units, piers are usually provided at entrance to guide the water and reduce the span of the roof slab.

According to the practice of the Bureau of Reclamation, the design of metal cases and concrete cases differs only slightly in principle. The metal cases are designed and furnished by the turbine manufacturer, whereas the structural design of concrete cases is furnished by the federal government. It is specified that engineers performing the hydraulic machinery design prepare the final neat-line drawings for use by the structural designers in the design of the concrete. The neat-

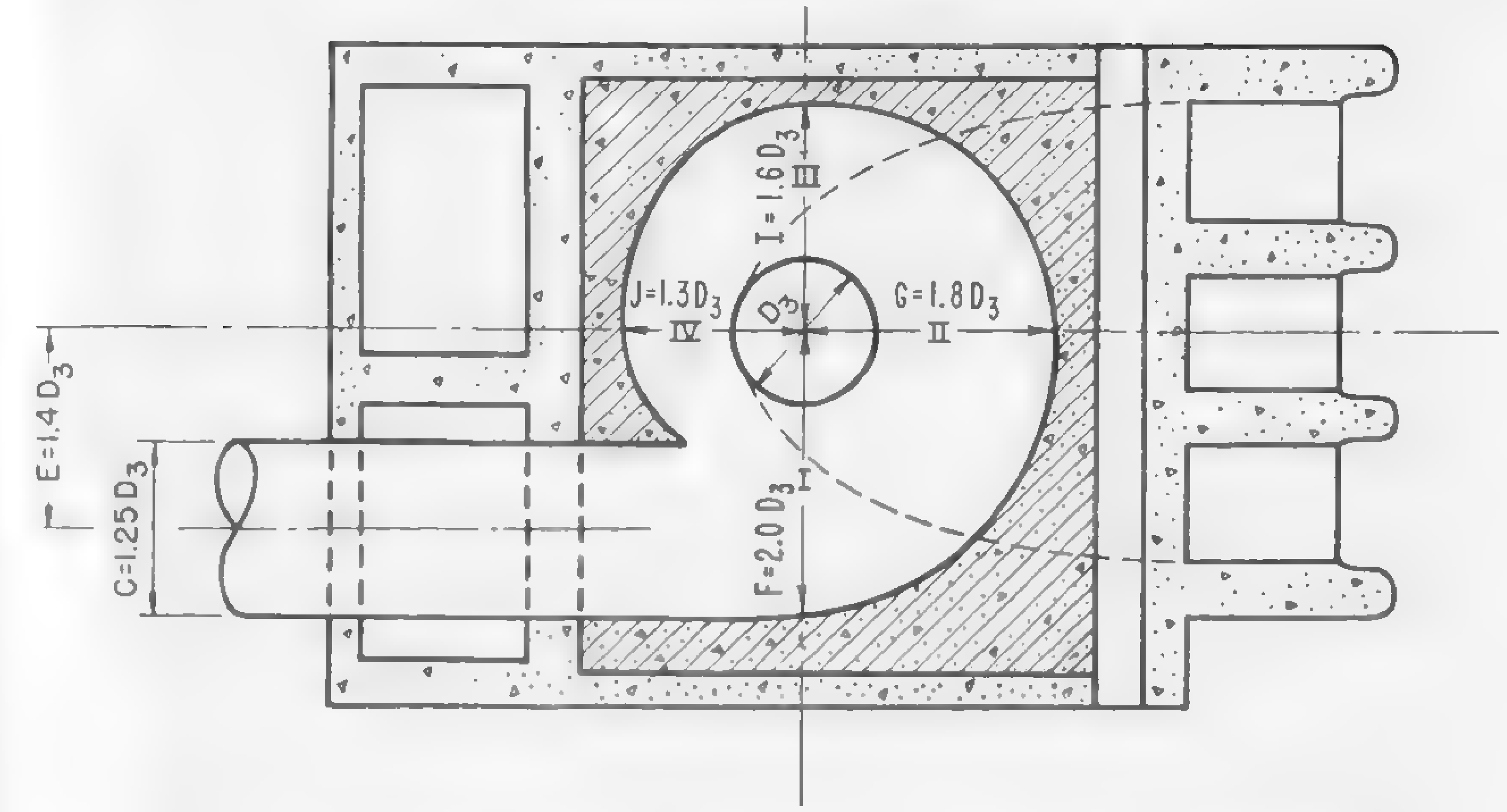
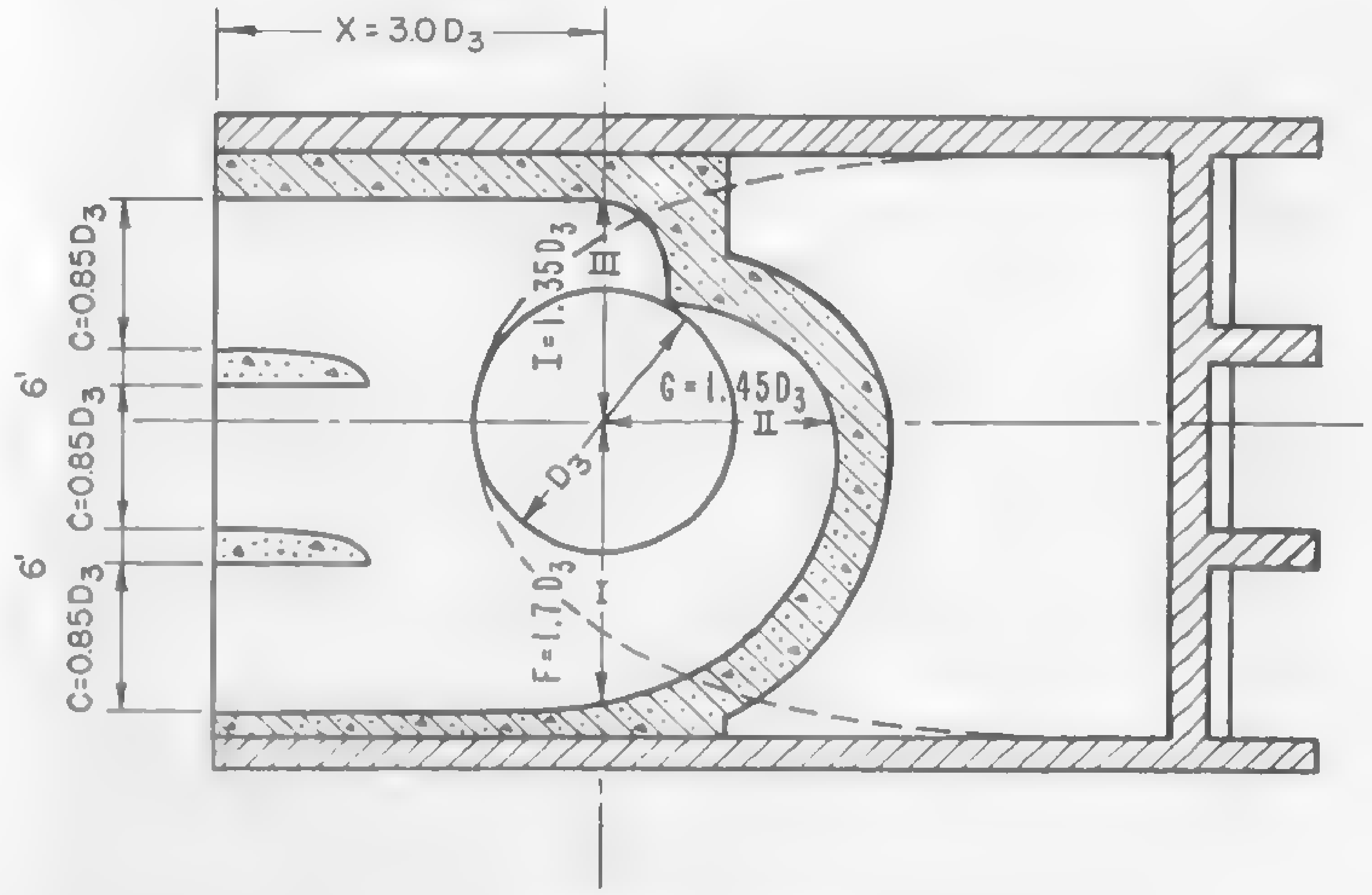


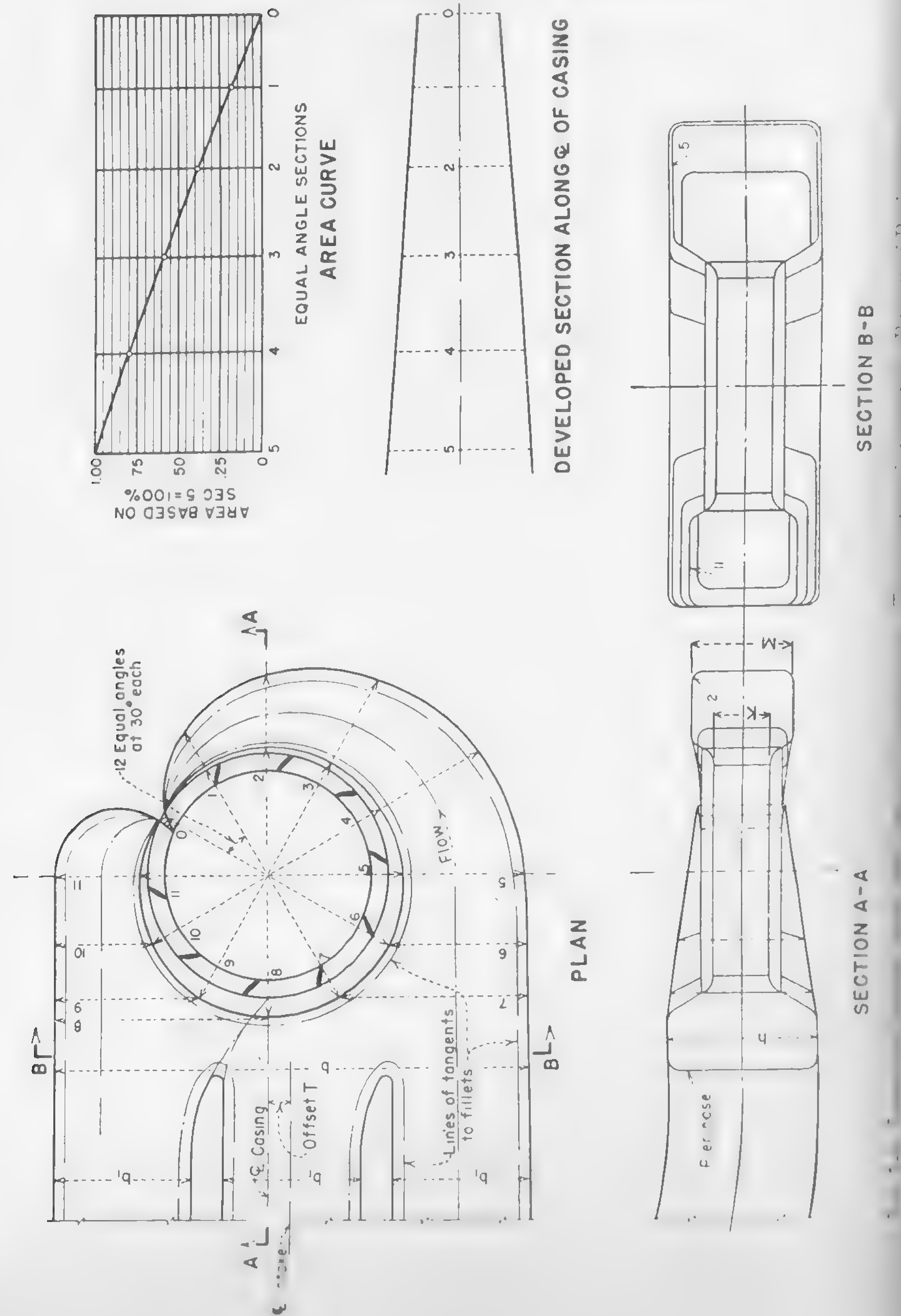
FIG. 5–3A. Approximate setting dimensions for preliminary estimates, Francis-type steel scroll.



Range of Values

$X : 2.0D_3 - 4.0D_3$ $\quad$ $G : 1.3D_3 - 1.7D_3$
$C : 0.76D_3 - 1.12D_3$ $\quad$ $I : 1.2D_3 - 1.5D_3$
$F : 1.2D_3 - 1.8D_3$

FIG. 5-3B. Approximate setting dimensions for preliminary estimates, propeller-type concrete scroll.



line drawings are developed from approximate outline drawings furnished by the turbine manufacturer. The stations selected for detailing are at a maximum of from 3 to 4 ft apart in order to facilitate interpolation of rib sections between these given sections, if this is required in the construction of the forms. Reference is often made to the quadrants of the scroll case. The positions of the quadrants are designated by the Roman numerals, I, II, III, and IV. Figure 5–3A shows the position of each quadrant for full scrolls built of metal, and Fig. 5–3B for semiscrolls, usually built of concrete. The dimensions, indicated as ratios with respect to D_3, merely express an average, determined from the detail drawings of a large number of plants already built. It will be noted that no Quadrant IV is indicated for semiscroll cases.

Figure 5–4 shows the general practice of the Bureau of Reclamation for the design of semiscrolls. Velocity and direction of flow dictate the area and position of the individual cross sections. It will be noted that each of the entrance channels is assumed to admit one third of the total flow to the turbine. This requires that the center line of the entrance channels be offset from the center line of the casing. The ends of the piers must also be shaped to properly divide the flow. As an example of the application of this practice Table 5–2 gives the scroll case dimensions of the Wheeler plant on the Tennessee River. This plant was designed by the Bureau of Reclamation and built by the Tennessee Valley Authority. It will be noted from Fig. 5–4 that the cross sections of the scroll case are symmetrical about the horizontal center line. This is not a fixed rule, as seen from the dimensions given for the Wheeler scroll.

TABLE 5–2

Dimensions of Scroll Case, Wheeler Project

Intake width C (Fig. 5–3B)		3 @ 18.00
Intake entrance above horizontal ℄		40.0
Intake entrance below horizontal ℄		10.0
Intake entrance total		50.0
Intake height h (Fig. 5–4):		
Inside of piers above horizontal ℄		19.28
Inside of piers below horizontal ℄		13.0
Inside of piers total		32.28
Plan dimension	F (Fig. 5–3B)	36.0
	G	28.24
	I	30.00
Height	K	9.35
	M	19.90
Offset T (Fig. 5–4)		3.00

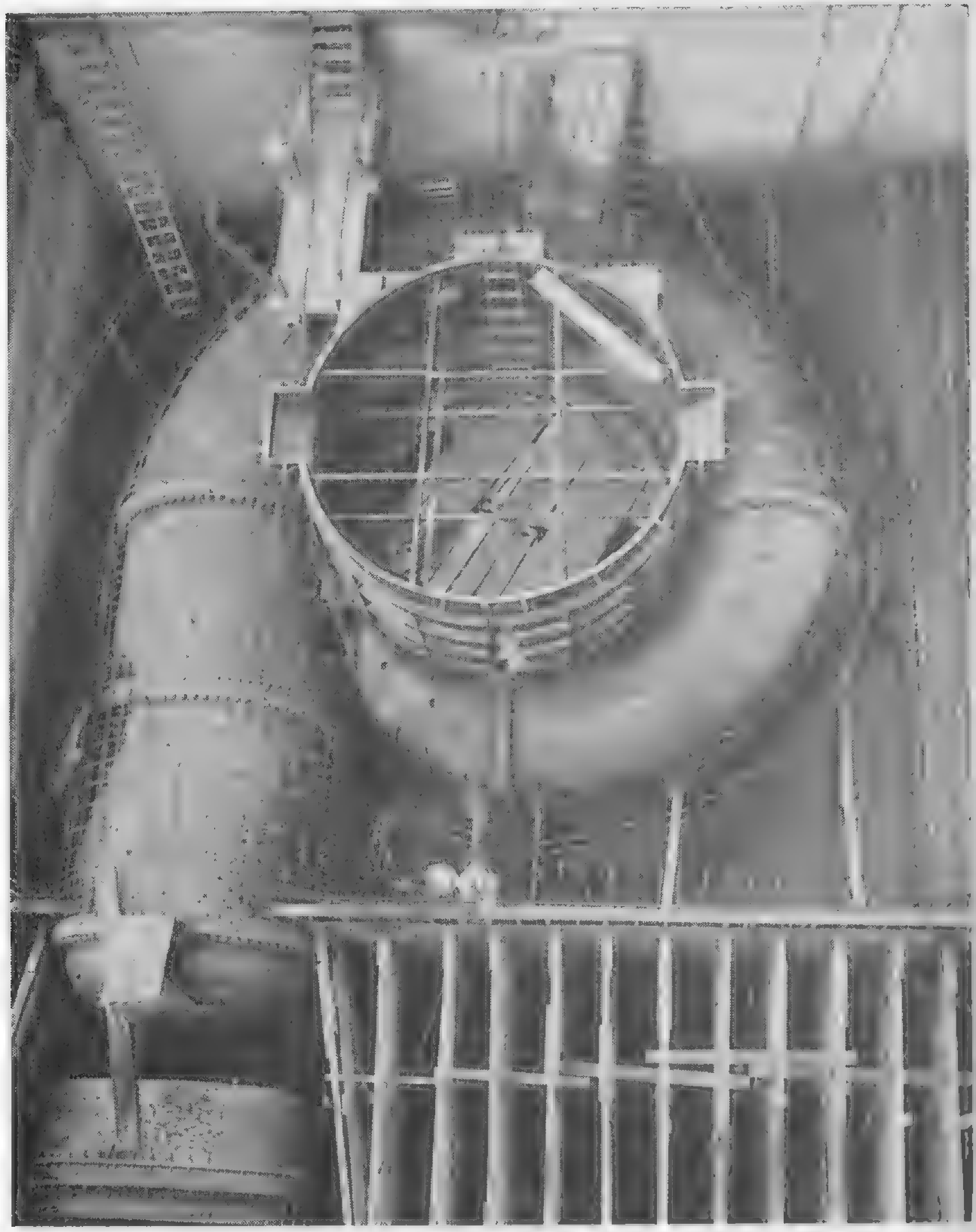
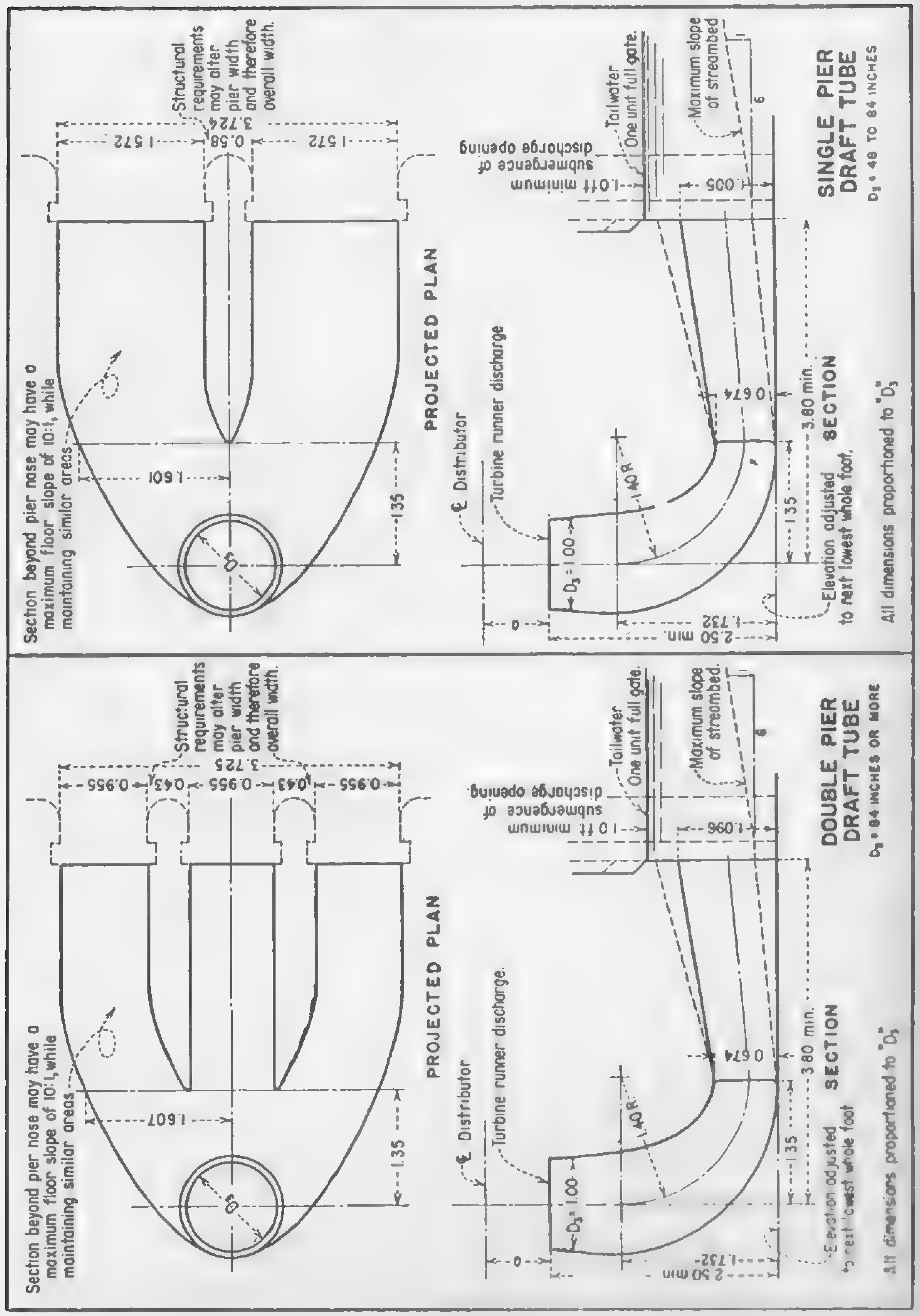


FIG. 5-5. Welded steel scroll case at Hungry Horse Dam. Inlet diameter 11 ft; rated head 400 ft. (Bureau of Reclamation)

5-7. The Draft Tube. For preliminary design it is only necessary to define the maximum depth, width, and length of the draft tube. The actual neat-line layouts of the final tube will be determined from designs furnished by the turbine manufacturers. Figure 5-6 gives the preliminary basic outlines of draft tubes suitable for reaction turbines. These outlines are to be used as a basis for preliminary layouts and for establishing limits to which it will be economically feasible to permit the turbine manufacturers to design the draft tubes.

Figures 5-7 and 5-8 show the optimum proportions of the two-pier elbow draft tube and one-pier elbow draft tube, respectively. The proportions are based on extensive laboratory development work and experience gathered from many outstanding plant installations of high turbine efficiency. They are acceptable for final designs where the turbine manufacturer prefers its use to an alternate design, or where it is necessary to construct the powerhouse substructure prior to the purchase of the turbines. In the latter case, a print of the detail drawing with area curves should be included in the turbine specifications.

The proportions can be considered as standard design except that certain variations in the lengths of the vertical section and horizontal diffuser are permissible. The horizontal length of the draft tube, and hence the outflow velocity, is determined to some extent by the design of the power plant substructure. The outflow velocity may vary from 4 to 8 fps with good efficiency. The proportions of the elbow between Sections III and XV are inviolate. They should be faithfully adhered to except to obtain smooth transitions of surfaces and areas as required by the remainder of the draft tube. A single-pier or a double-pier draft tube sometimes may be used on runners of from 4 ft to 7 ft discharge diameter for structural reasons, but for hydraulic reasons piers should be avoided if possible. For runners with discharge diameters of 4 ft or less, either the standard elbow draft tube with no piers or the straight 12-degree conical draft tube may be selected. However, for runners with discharge diameters of less than 2 ft, the elbow type is seldom found to be economical. For the smaller sizes a conical extension between the runner discharge diameter and the draft tube outlet may be expedient. From the viewpoint of hydraulic efficiency, the conical draft tube should be used, but usually it is not the most economical type. In the case of station-service units adjacent to larger units, where the choice is not affected by economical considerations, the conical type would be the best choice. Figure 5-9 shows the results of studies on 58 recently designed plants to determine ratios of dimensions to discharge diameter D_3. It will be noted that values in Figs. 5-7 and 5-8 are in relatively close agreement with those in Fig. 5-9.



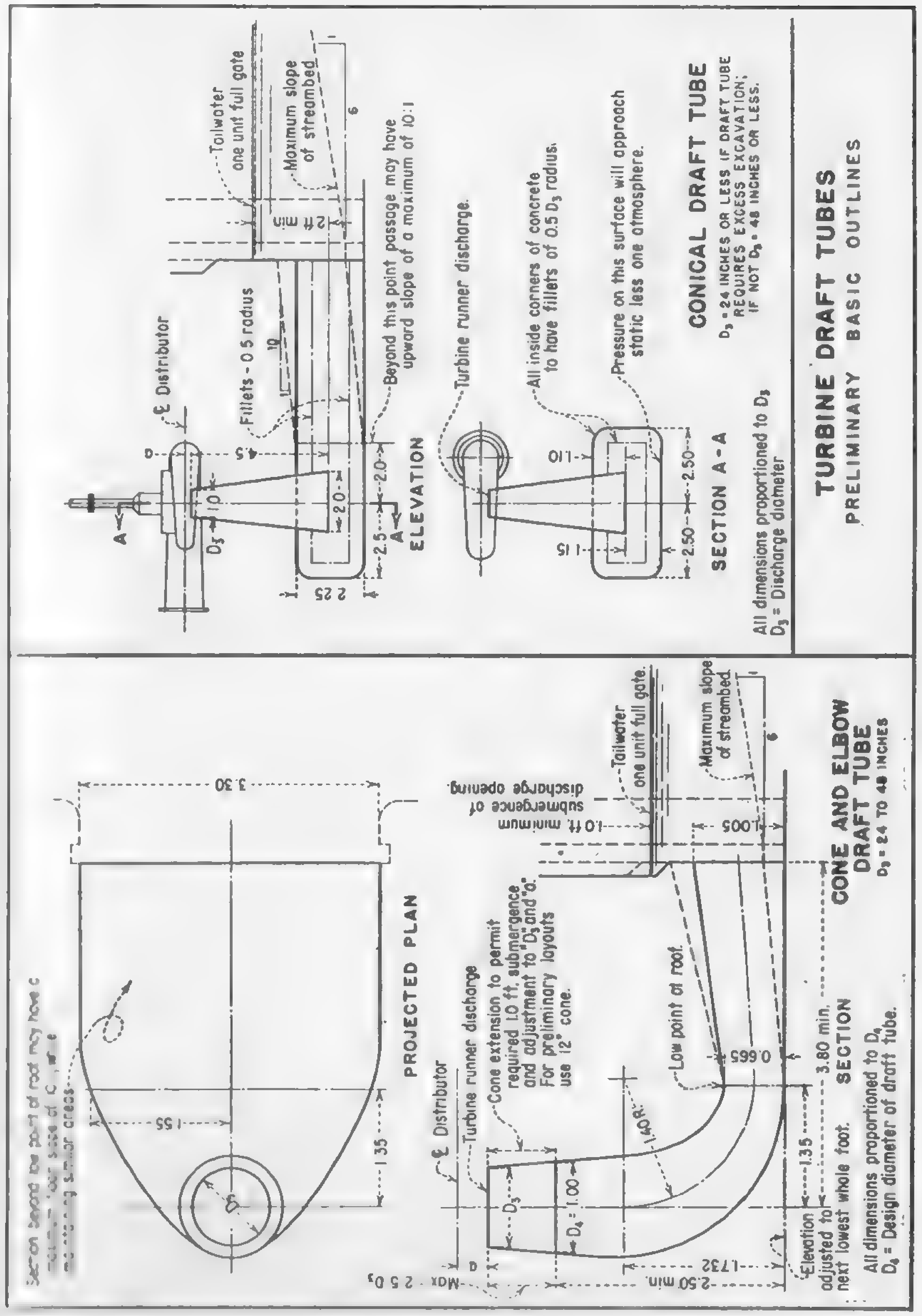
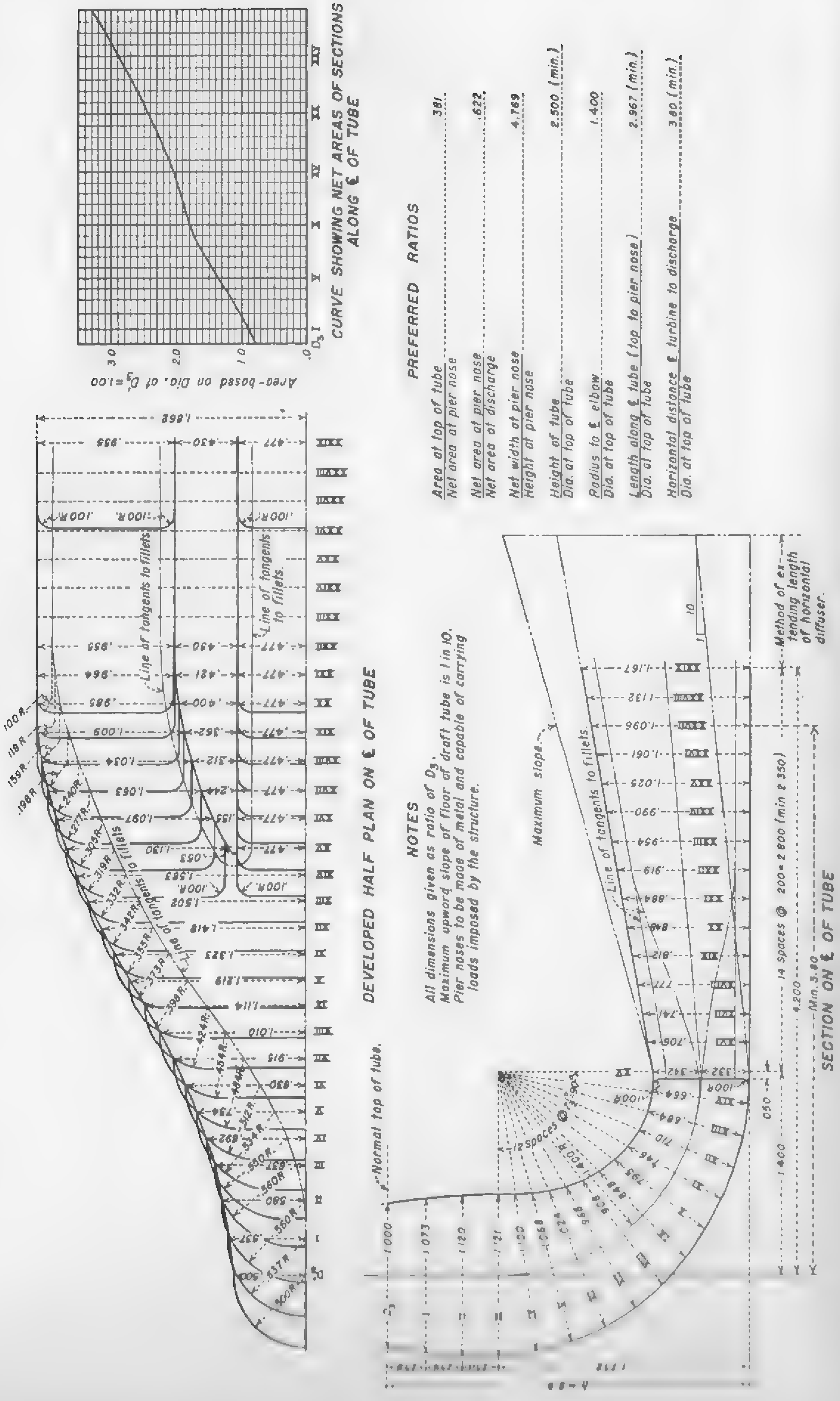
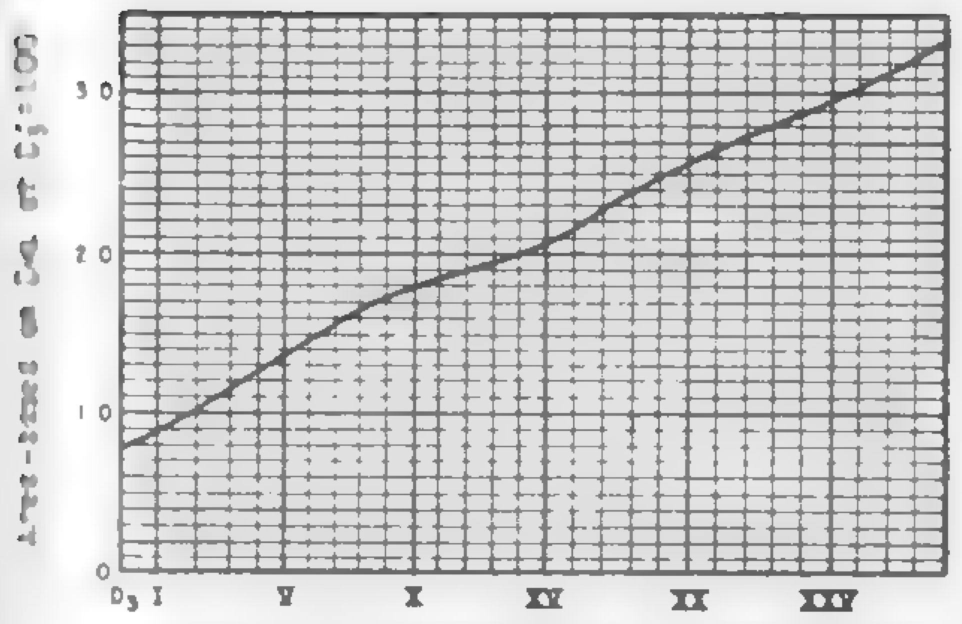


FIG. 5-6. Basic types of reaction turbine draft tubes. (Bureau of Reclamation)



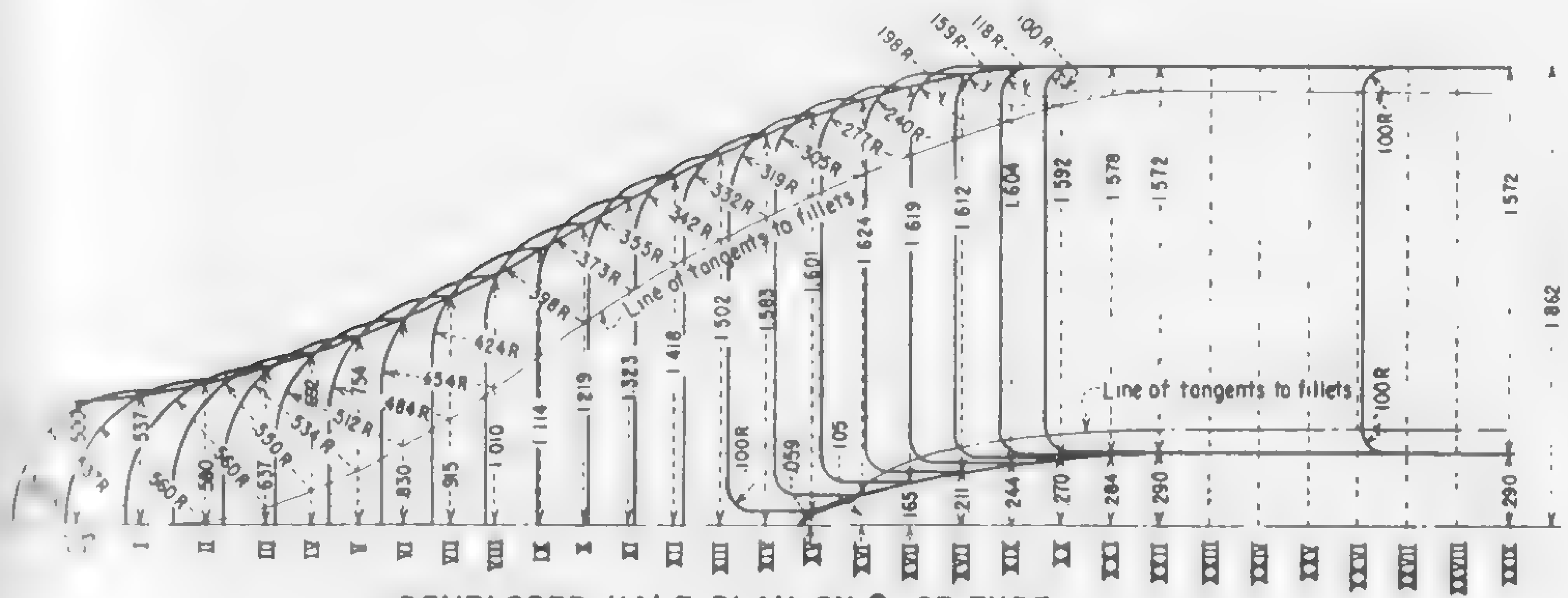


CURVE SHOWING NET AREAS OF SECTIONS ALONG ℄ OF TUBE

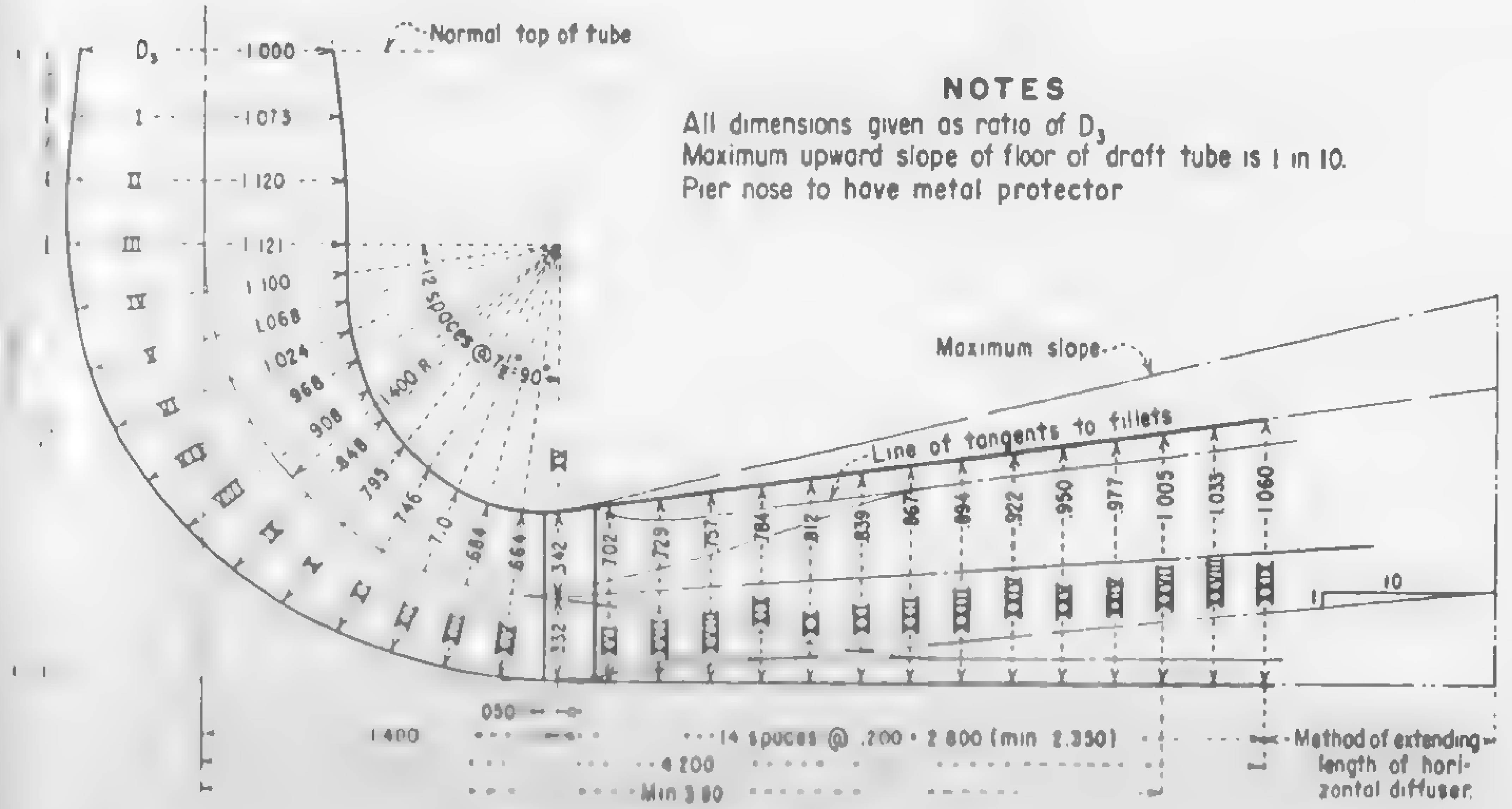
PREFERRED RATIOS

Area at top of tube / Net area at pier nose	.381
Net area at pier nose / Net area at discharge	.622
Net width at pier nose / Height of pier nose	4.769
Height of tube / Dia. at top of tube	2.500
Radius to ℄ elbow / Dia at top of tube	1.400
Length along ℄ tube (top to pier nose) / Dia. at top of tube	2.967 (min.)
Horizontal distance ℄ turbine to discharge / Dia at top of tube	3.80 (min.)

TURBINE DRAFT TUBES - ONE PIER
OPTIMUM PROPORTIONS



DEVELOPED HALF PLAN ON ℄ OF TUBE

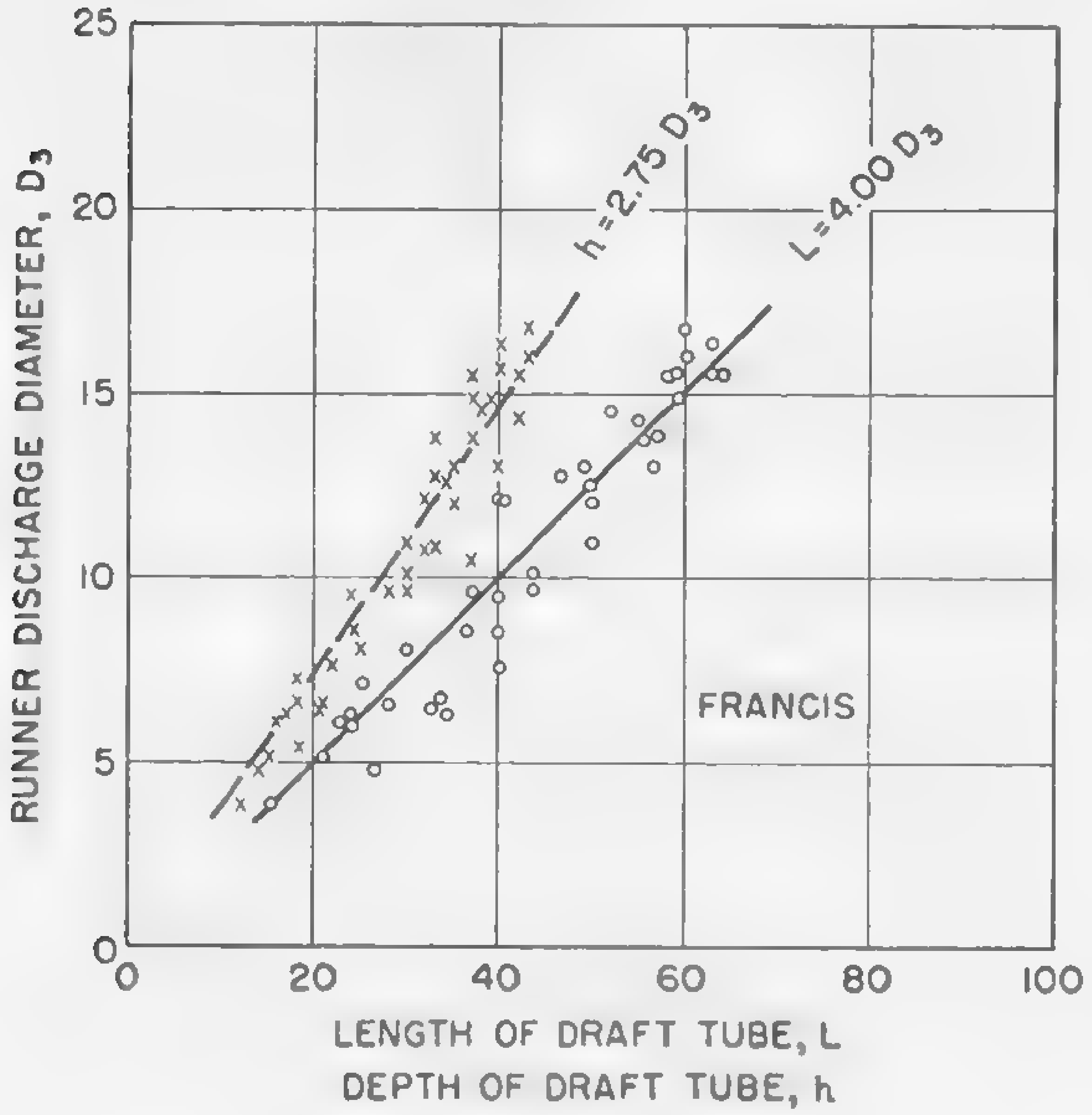


NOTES

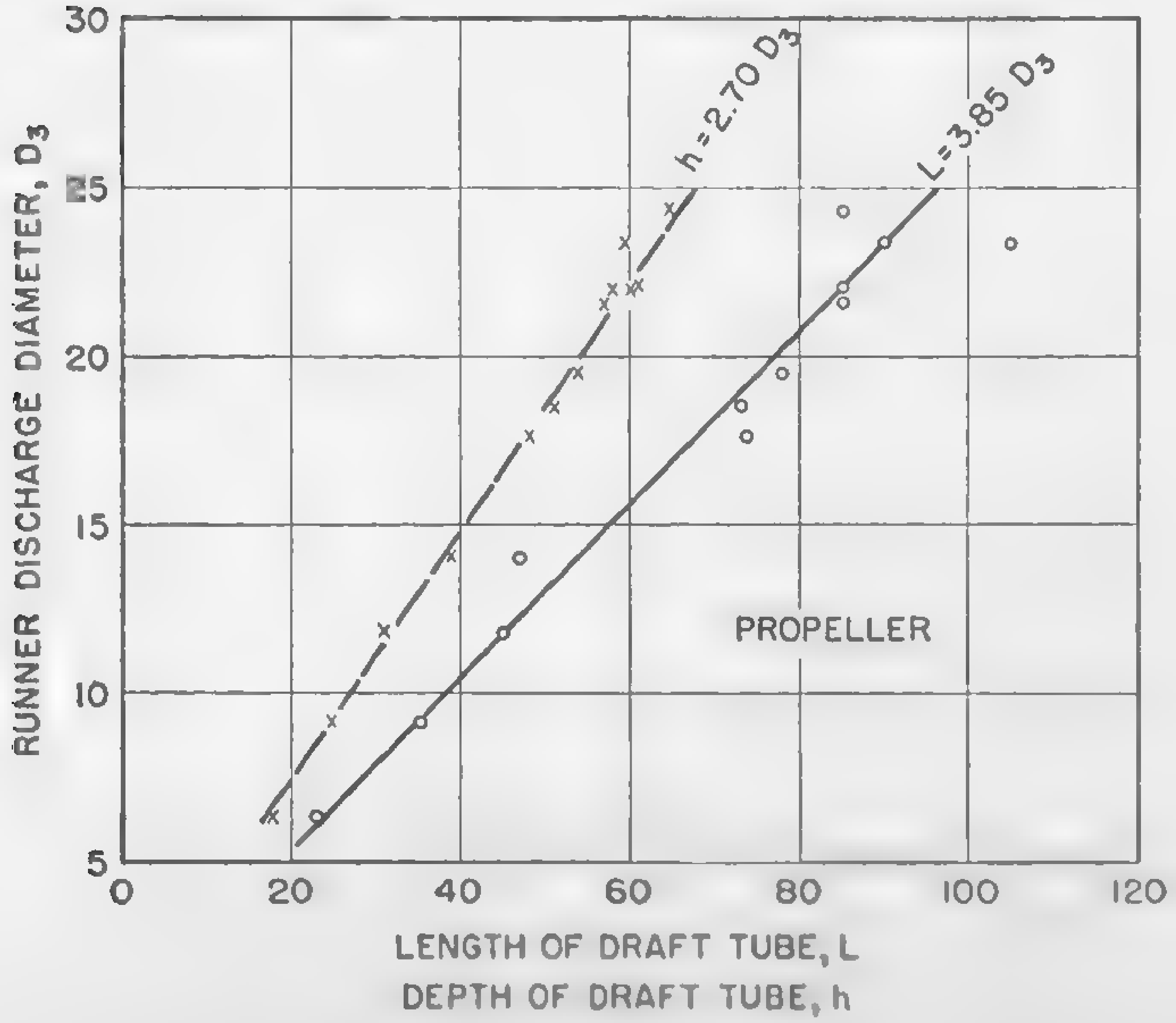
All dimensions given as ratio of D_3.
Maximum upward slope of floor of draft tube is 1 in 10.
Pier nose to have metal protector

SECTION ON ℄ OF TUBE

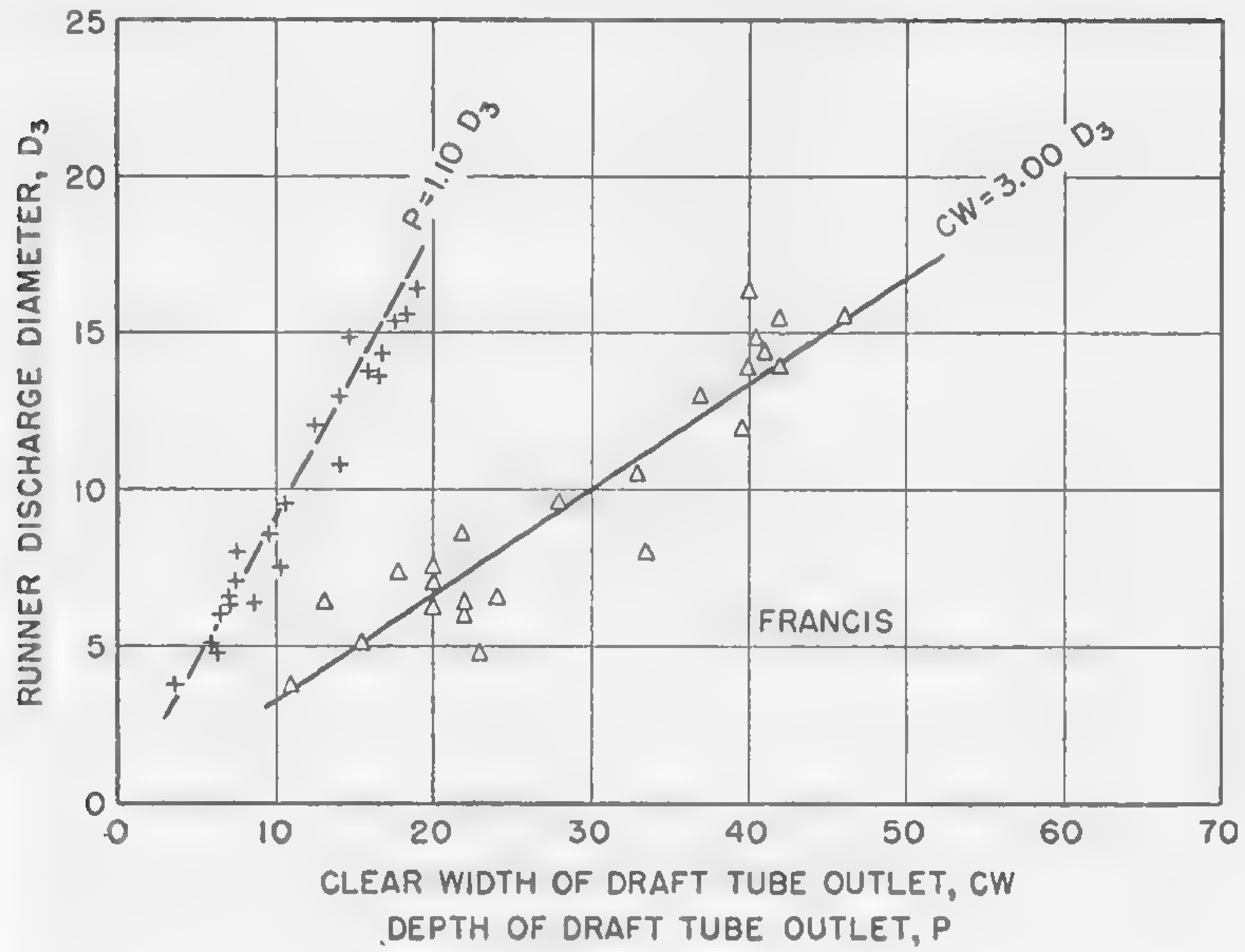
Fig. 5.8 Proportions for single-pier draft tubes. (Bureau of Reclamation)



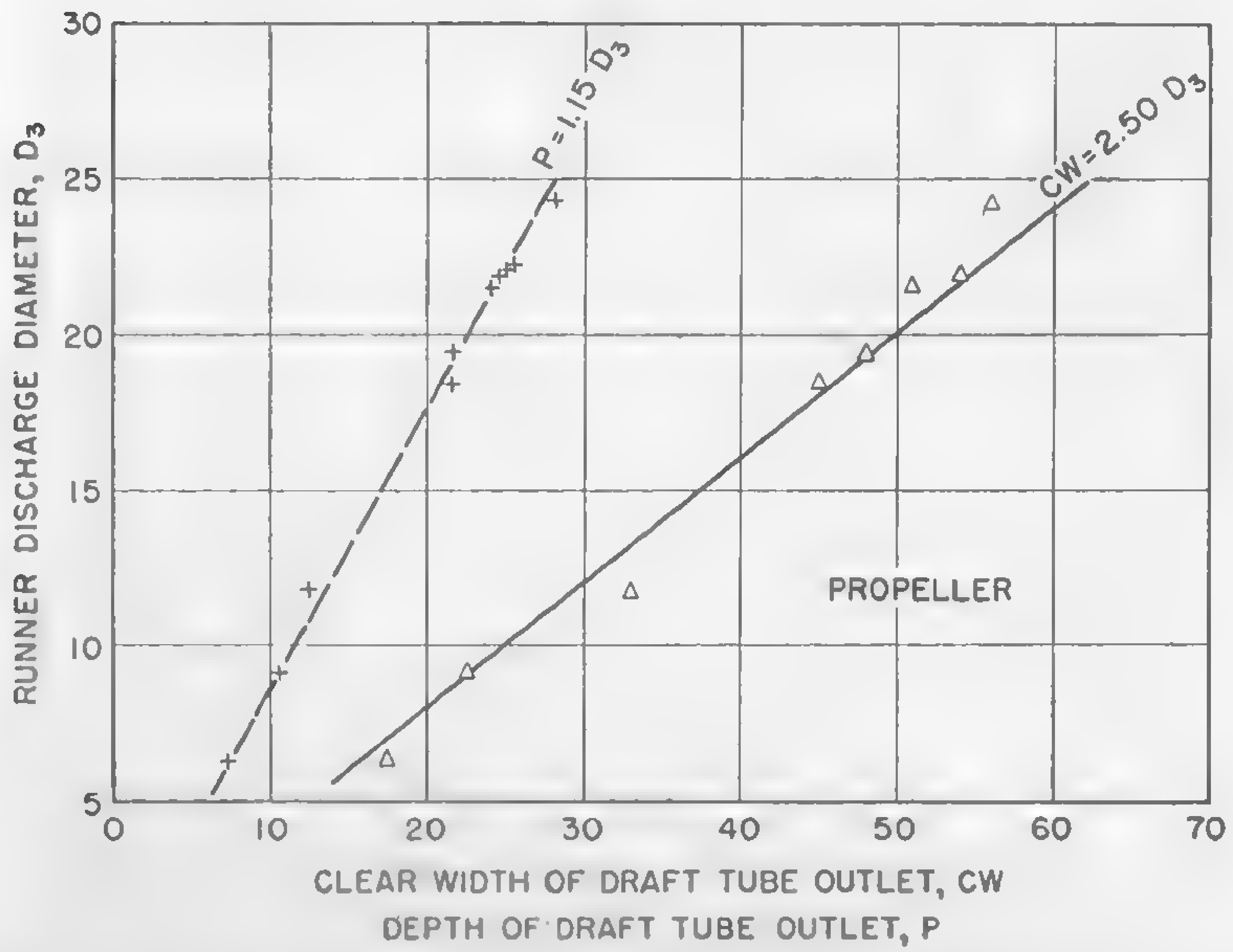
(A)



(B)



(C)



(D)

Fig. 5-9. Ratios of length (A), depth (B), outlet width (C), and height (D) to discharge diameter. Based on author's study of 58 plants.

As mentioned in Art. (4–7), cavitation may take place in the draft tube. The usual method of resisting erosion and pitting from possible cavitation is to provide plate-steel liners below turbine runners. The length of the liner depends on the type of runner and the head. For low-head Francis turbines, these liners should extend not less than a distance of D_3 below the point of attachment to the turbine bottom ring under the most favorable conditions.

For axial-flow turbines and for heads up to 350 ft, the liner may be terminated at a point where the area of the draft tube is twice the area at D_3. For heads over 350 ft, the liner should extend past the elbow section to include the pier noses as an integral part of the liner. The liner should be of rugged construction and firmly anchored to the concrete at frequent intervals to prevent buckling from external water pressure caused by seepage between the liner and the concrete. The design value of this external pressure for turbines having metal draft-tube liners results from the maximum tailwater with the unit dewatered. For turbines having concrete cases, it is the full base pressure. The same design conditions should be assumed in computing the external pressure on the turbine-pit liner.

5–8. Powerhouse. The powerhouse type may be classified in two ways: (1) according to the method of housing the main generator units and (2) according to structural considerations. The types according to housing are indoor, semi-outdoor, and outdoor. In the indoor type, the generating machinery is fully enclosed by a building of sufficient height to permit handling and transfer of equipment by means of an indoor crane. In the semi-outdoor type, the generator room is fully enclosed, but the equipment is handled by an outdoor gantry crane through hatches in the roof. In the outdoor type, there is no generator room, but the generators are housed in individual cubicles on or recessed in the roof deck. The equipment is handled by a gantry crane.* The selection is determined by economic considerations, climatic conditions, and convenience in operation. Figure 2–11 shows an indoor type; Fig. 2–12, an outdoor type; and the top view of Fig. 5–11, a semi-outdoor type. In Figs. 5–12 and 5–13 the types are:

Indoor: Norris, Ocoee, Pickwick Landing, Apalachia, Chickamauga, Fontana, and Guntersville
Semi-outdoor: Kentucky, Watts Bar, Cherokee, Fort Loudoun
Outdoor: Hiwassee

* U. S. Corps of Engineers, *Manual of Civil Works Construction* (Washington, D. C., 1948), Part CXXX, Chap. 1.

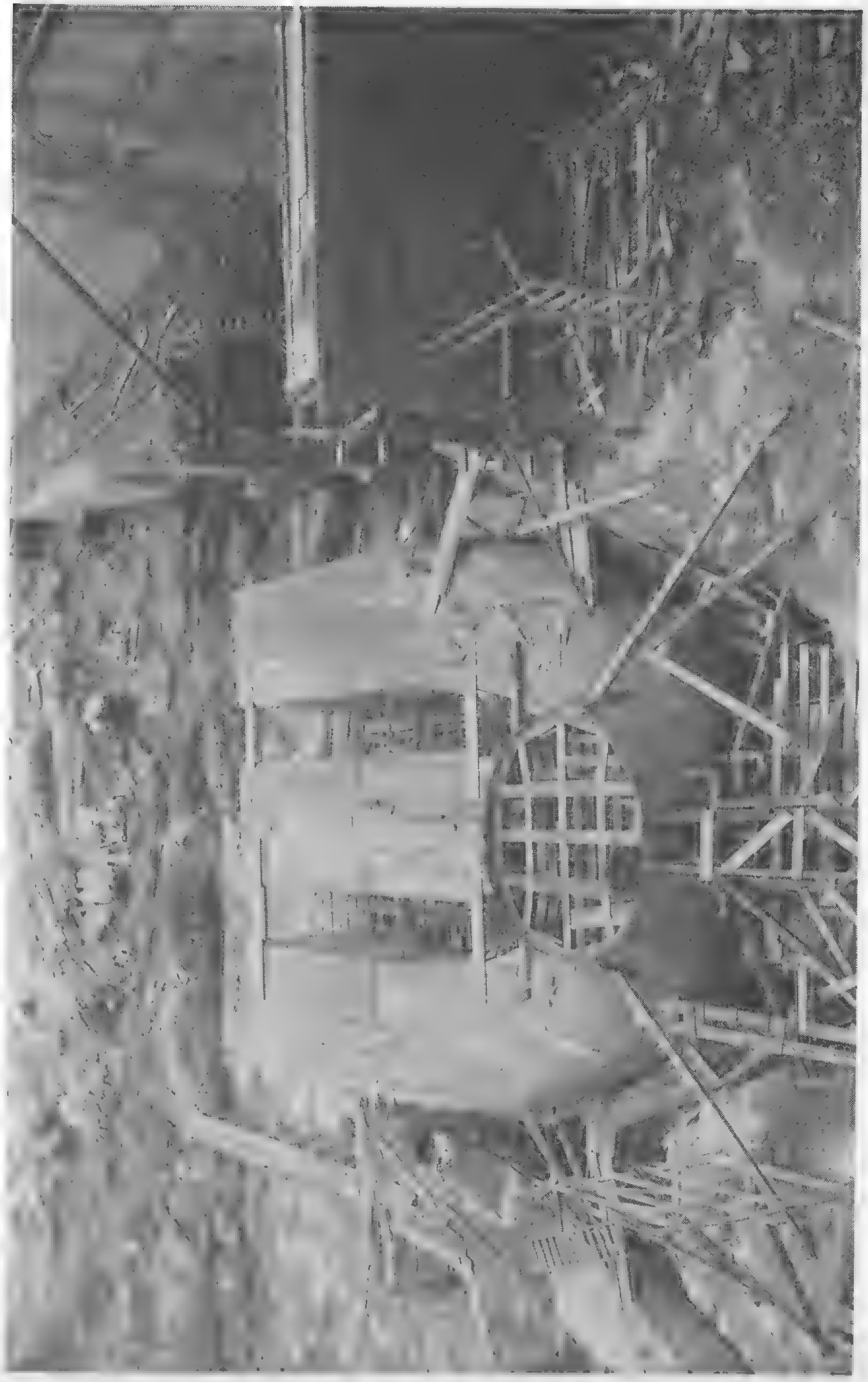


Fig. 5-10. Draft tube forms for the Norris Dam powerhouse. (Tennessee Valley Authority)

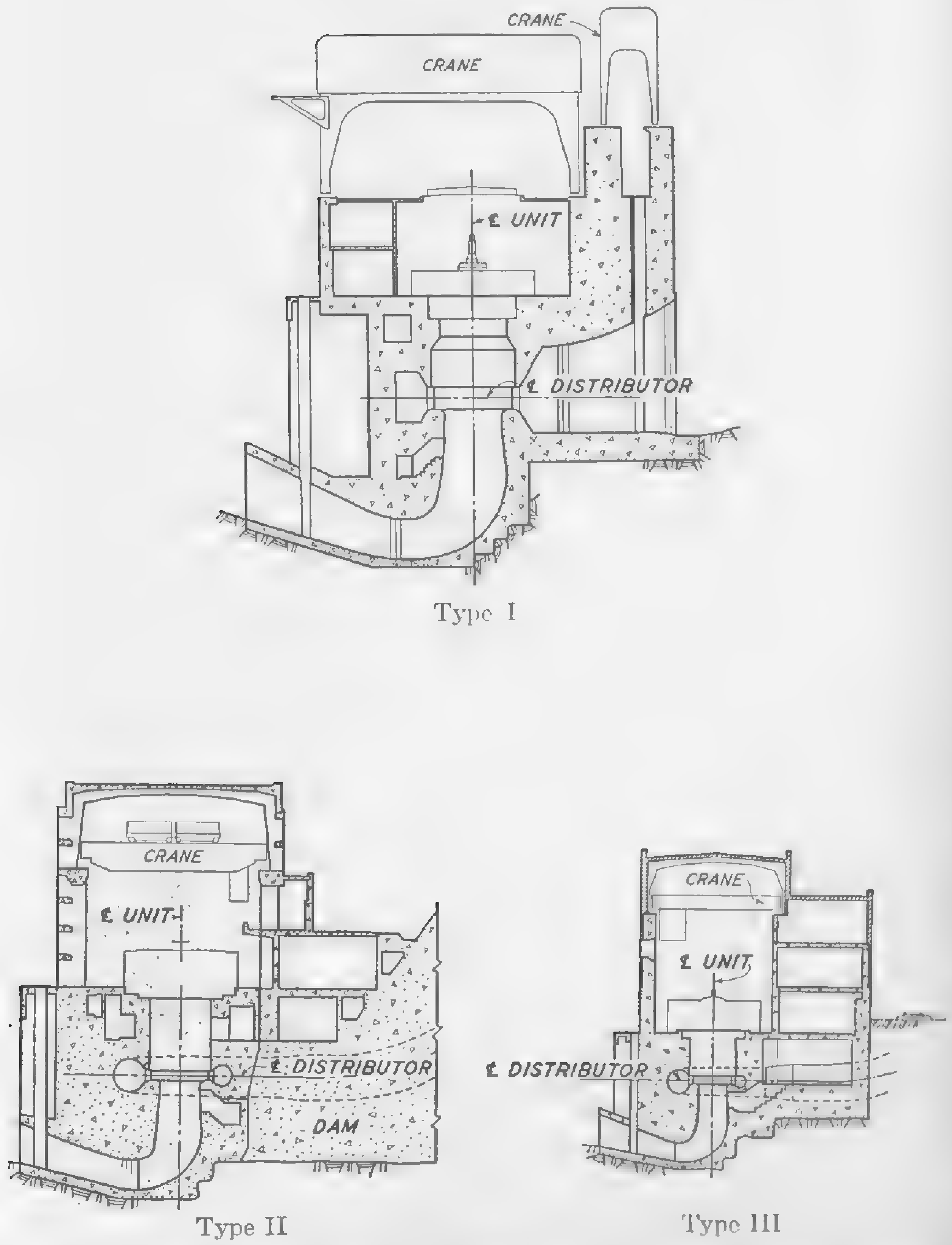


FIG. 5-11. Types of powerhouses, TVA classification. (Tennessee Valley Authority)

Structurally the Tennessee Valley Authority classifies powerhouses according to three types (Fig. 5-11): *

Type I. The powerhouse functions as a dam. The unit intake and unit substructure usually are integrated into one block to resist water load. The lengths of the blocks or bays should correspond with the lengths of the monoliths in the dam and should not exceed 60 ft except for low-head projects of large capacity where the length may be increased to 90 ft.

Type II. The powerhouse does not assist in carrying headwater load. It is located at the toe of a concrete gravity bulkhead with conduits leading through the dam space between unit substructure and downstream face of the dam is usually utilized for station auxiliary space.

Type III. The powerhouse is entirely removed from the dam. A penstock carries water into the unit substructures.

Figures 5-12 and 5-13 show details of specific TVA plants.

The length of the substructure is usually controlled by the widths of the draft tubes and piers and the erection bay. The distance between units varies between $3.5D_3$ and $6D_3$. Tables 5-2 and 5-3 give values for certain recent TVA installations. Allowances for working space, or erection bay approximately equal to the area occupied by one unit should be made in establishing the length of the powerhouse. The possible or probable addition of future units should always be kept in mind in a consideration of substructure dimensions.

The width of the powerhouse substructure usually depends upon the length of the scroll case and draft tube. It is difficult to establish ratios between the width of the substructure and the discharge diameter, but the following ratios will give approximate dimensions.

For propeller-type runners, the width of substructure equals about 7.5 to 8.0 times the runner discharge diameter. For Francis-type runners, the width is about 6 times the runner discharge diameter.

The powerhouse superstructure layout must be designed to accommodate the generating units, erection bay, service areas, and public areas. A breakdown of these four general spacings is indicated in Table 5-3. The data shown in Table 5-3 were derived from a study by the author, and are based upon 12 plants constructed by the Tennessee Valley Authority. The results cannot be considered as standard for all hydroelectric developments. For example, studies made on 18 plants by the Corps of Engineers indicated that the ratio between the area of the

* *Design of TVA Projects*, Vol. I, *Civil and Structural Design*, TVA Technical Report No. 21 (Knoxville, Tenn.: Tennessee Valley Authority, 1952), p. 181.

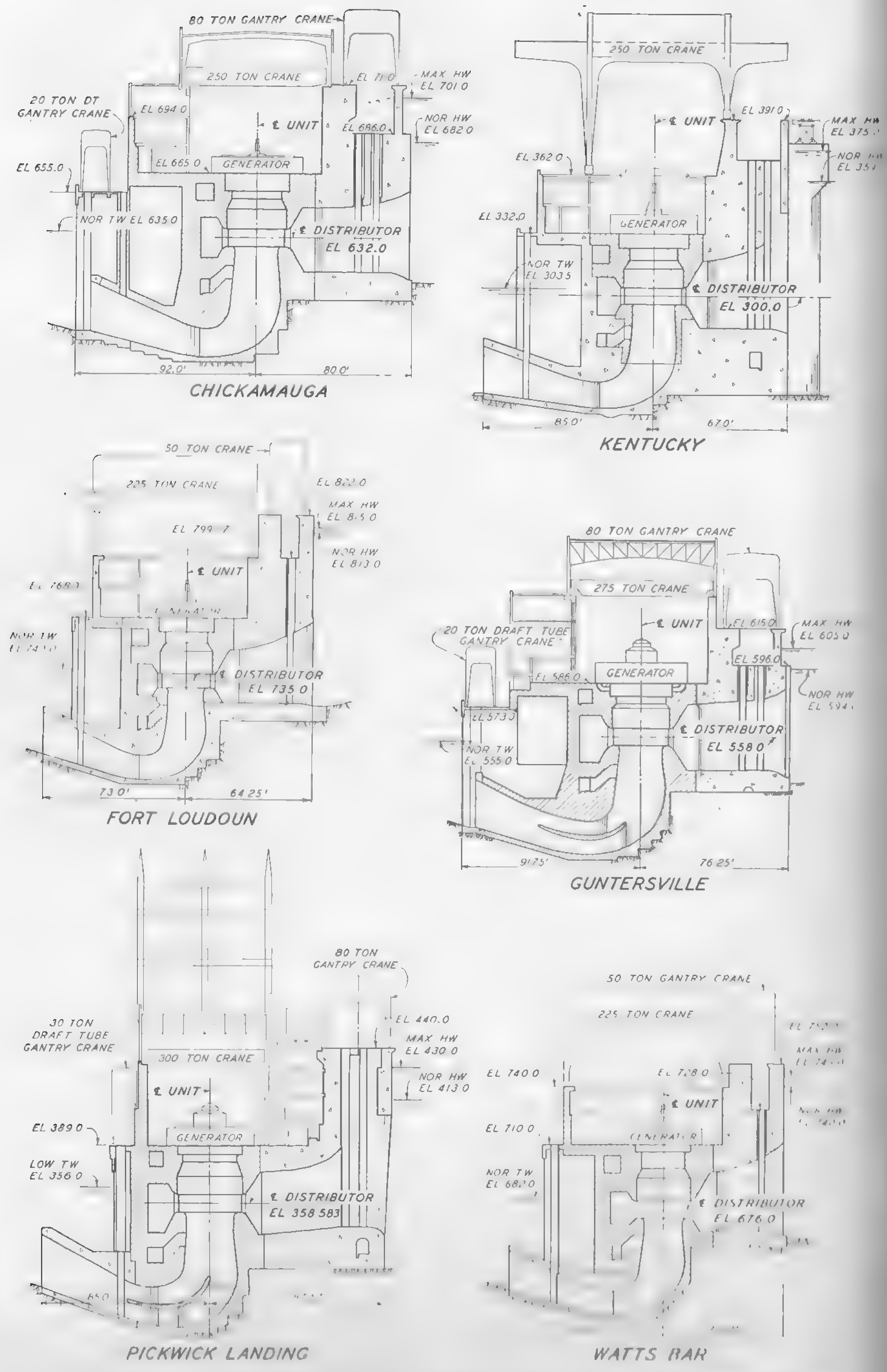


FIG. 5-12. TVA installations with propeller-type runners. (Tennessee Valley Authority)

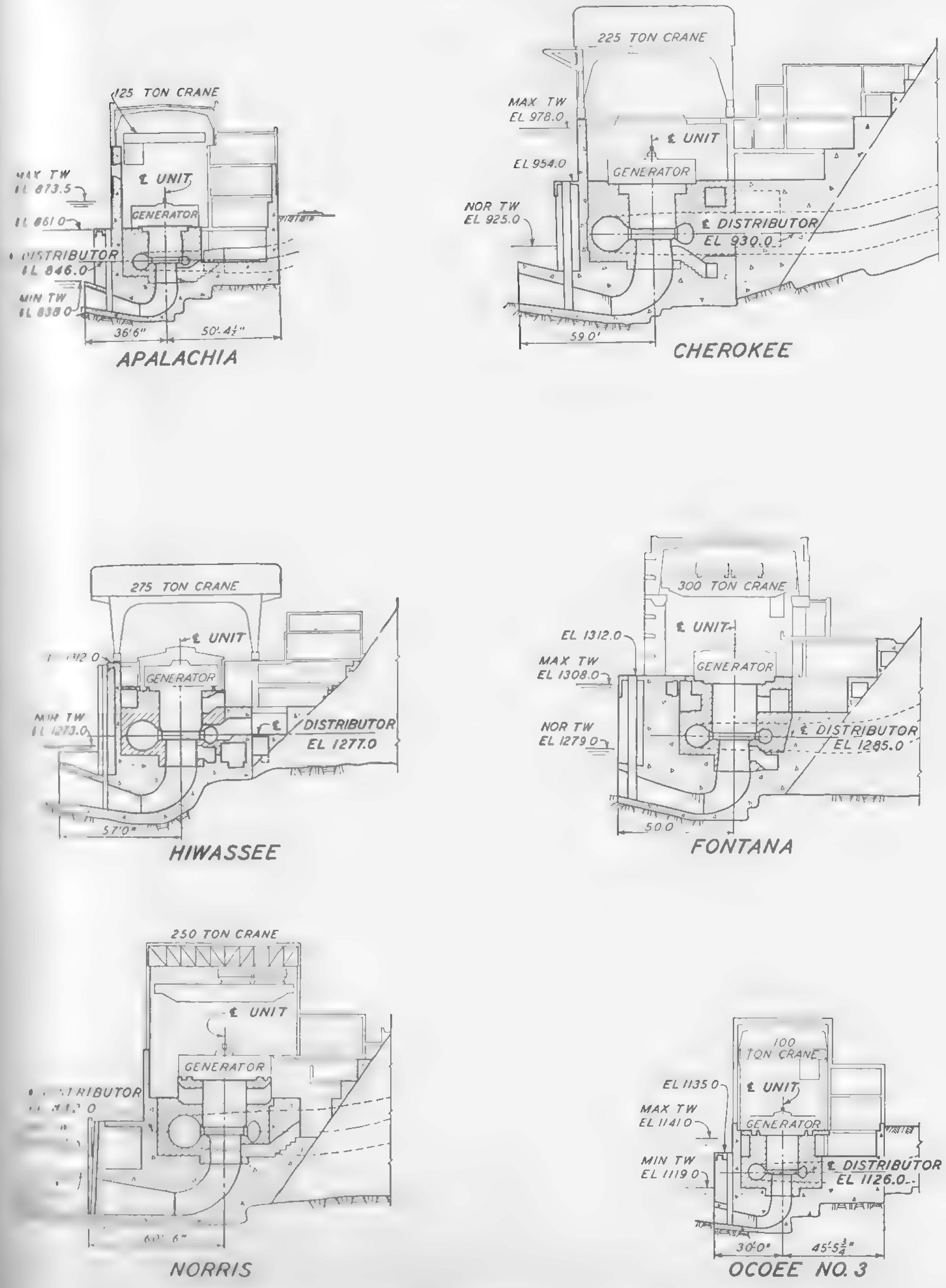


FIG. 5-13. TVA installation with Francis-type runners. (Tennessee Valley Authority)

generator room and the area of the space occupied by the generator is 2.98, whereas for 13 plants designed by the Bureau of Reclamation the ratio is 3.98. The range in the ratios is as follows:

	Range	*Average*
TVA	2.36 to 3.82	3.14
Corps of Engineers	2.33 to 3.60	2.98
Bureau of Reclamation	2.60 to 5.54	3.98

The over-all average of the ratio for the 43 plants is 3.33.

The actual diameters of the generators were used in these studies. For preliminary estimates the approximate diameter and hence the area of the generator can be determined from Fig. 5–14.

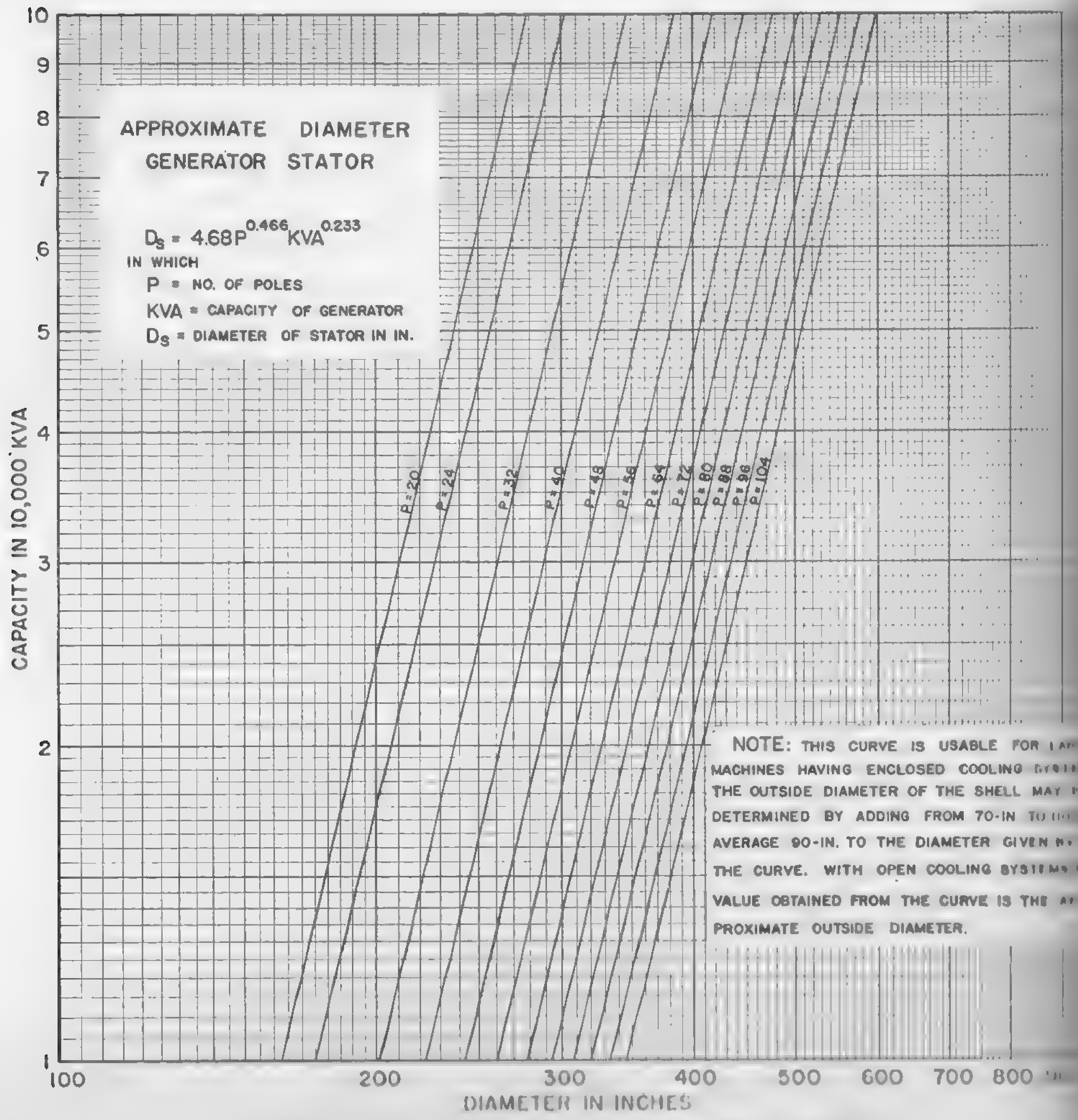


Fig. 5-14. Diagram for determining generator diameter.

TABLE 5–3

Approximate Superstructure Space Requirements
(Based on TVA installations)

Facility	Space
Generating Units	
Generator area per unit	3.15 times the area occupied by the outside shell of the generator
Other generator facilities, including exciter, unit boards and switch gear, cable and general lead galleries, pipe gallery, scroll case and draft tube access valve room, and lower gallery	These spaces average approximately 72 per cent of the generator room area and range from 49 to 98 per cent. They are usually contained within and beneath the generator room and therefore do not add to the over-all length and width of the powerhouse
Operating areas, erection bay	Variable depending upon location of access railways or highways, type of powerhouse, etc. A general rule would be to provide an erection area equal to from 75 to 150 per cent of the area of the generator room.
Machine shop and tool room	Average 400 sq ft per main generating unit. Range 180–620 sq ft.
Electrical shop	Average 125 sq ft per main generating unit. Range 56–190 sq ft.
Storage, bar, and small parts	Average 300 sq ft per main generating unit. Range 120–480 sq ft.
Laboratory and dark room	Average 340 sq ft. Total per plant.
Control: Includes control room, terminal room, and spreading room	Average 1065 sq ft per main generating unit. Range 670–1375 sq ft.
Control room	Average 560 sq ft per main generating unit if not otherwise provided for.
Spreading room	Average 400 sq ft per main generating unit if not otherwise provided for.
Terminal room	Average 350 sq ft per main generating unit if not otherwise provided for. For one- and two-unit plants space for control facilities may be provided for in other areas and do not require additional space.
Auxiliary mechanical equipment, including actuator cabinet and pressure tank, oil handling, oil storage, water treatment, raw-water, air compressor and receivers, CO_2 equipment, sump, and unwatering pumps	Average 1290 sq ft per main generating unit. Range 900–2100 sq ft.

(Continued on following page)

TABLE 5-3 (*Continued*)

APPROXIMATE SUPERSTRUCTURE SPACE REQUIREMENTS

(Based on TVA installations)

Facility	Space
Fan room and air conditioning	350 to 900 sq ft per main generating unit depending upon climate, public interest, and generator cooling requirements.
Auxiliary electrical equipment, including batteries, motor generator sets, auxiliary boards, station service transformers, gas electric set, and telephone room	Average 1180 sq ft per main generating unit. Range 870–1550 sq ft.
Employee facilities, including offices, assembly room, washrooms and toilets, lockers, kitchens, first aid and guard room; closets, janitor equipment, and overlooks	Depend upon location of plant with respect to accessibility to visitors and functional requirements. Must be based on experience. TVA provisions range from a total of 0 at Apalachia, and Ocoee No. 3 to 5380 sq ft at Chickamauga. Average for 10 plants 3520 sq ft, toilets and rest rooms vary from 0 to 810 sq ft.
Multi-purpose areas, including corridors, stairs, elevators, and inclines	Average 1150 sq ft per main generating unit. Range 370–1600 sq ft per main generating unit. A widely variable component depending upon powerhouse arrangement and type.

Figure 5–15 shows a view of the Trenche Development on the St. Maurice River in Quebec Province, Canada. It was designed and constructed in 1952 by the Shawinigan Engineering Company Limited for the Shawinigan Water and Power Company. The penstocks shown on the left are of fabricated steel plate and are 20 ft in diameter. The log chute in the middle is 14 ft wide and of rectangular cross section. The control gates to the right of the log chute are of the fixed-roller type, 21 ft wide and 22 ft high. The four spillway gates on the right are also of the fixed-roller type, 50 ft wide and 31.5 ft high. The total spillway capacity is 153,000 cfs. The dam is 1500 ft long and 235 ft high above the bed of the river. The outdoor powerhouse, approximately 60 ft by 384 ft, contains five units, each developing 65,000 hp at a rated head of 159 ft.

FIG. 5-15. Trenche Development, St. Maurice River, Quebec Province, Canada. (Shawinigan Water and Power Co.)

Figure 5-16 shows a view of the Bagnell development on the Osage River, Missouri. It was built by the Stone and Webster Corporation for the Union Electric Company of Missouri. The outdoor-type powerhouse contains six units of 33,500 hp, each operating under a head of 90 ft. The length of the powerhouse is 511 ft and the width from face of dam to outlet of draft tube is 150 ft. The total length of the dam is 2543 ft and the length of the spillway section is 520 ft. The spillway is controlled by 12 tainter gates, each 22 ft high and 34 ft long.

CHAPTER 6

APPURTENANCES FOR HYDRO PLANTS

6–1. Introduction. The construction of a hydroelectric power plant requires the use of some or all of these appurtenances or accessories in connection with its operation: dams, control works, gates, waterways, penstocks, tunnels, fishways, pressure relief valves, surge tanks, cranes, turbine governors, transformers, switching equipment, and transmission lines.

The civil engineer is directly concerned with the location and design of many of these accessories, but only indirectly with others.

6–2. Dams. Except for brief mention in connection with hydroelectric projects, the design and construction of dams is outside the scope of this book. The student is referred to the literature and to courses in the curriculum relating to this subject.

6–3. Control Works. Structures and devices to control the supply of water for hydroelectric plants fall somewhat in the same category as dams. Actual design procedures may be found in special treatises devoted to gates and valves. The principal elements of control works consist of gates of various types and the structures necessary for their operation. The control works may also include devices for the protection of the gates and the hydraulic machinery. These protective devices may include trashracks, log and ice booms, debris clearing devices, and heating elements for the avoidance of ice troubles.

6–4. Gates. The types of gates which are used in connection with hydroelectric developments are: drum gates, radial gates, vertical lift gates, roller gates, caterpillar gates, etc.

The *drum gate* (Fig. 6–1) is an automatic type used to control the surface elevation of the water in a reservoir. It consists of a watertight drum built of properly braced plate steel. The drum is hinged on the upstream side and floats in a water-filled concrete chamber. The level of the water in the chamber is controlled by a float which actuates a balanced valve. A rising level releases water from the float chamber, and a lowering level admits water to the chamber. The floating drum lowers or rises to increase or decrease the flow over the top of the drum.

The contour of the upstream face of the watertight chamber conforms with the ogee crest outline. This feature insures a high discharge

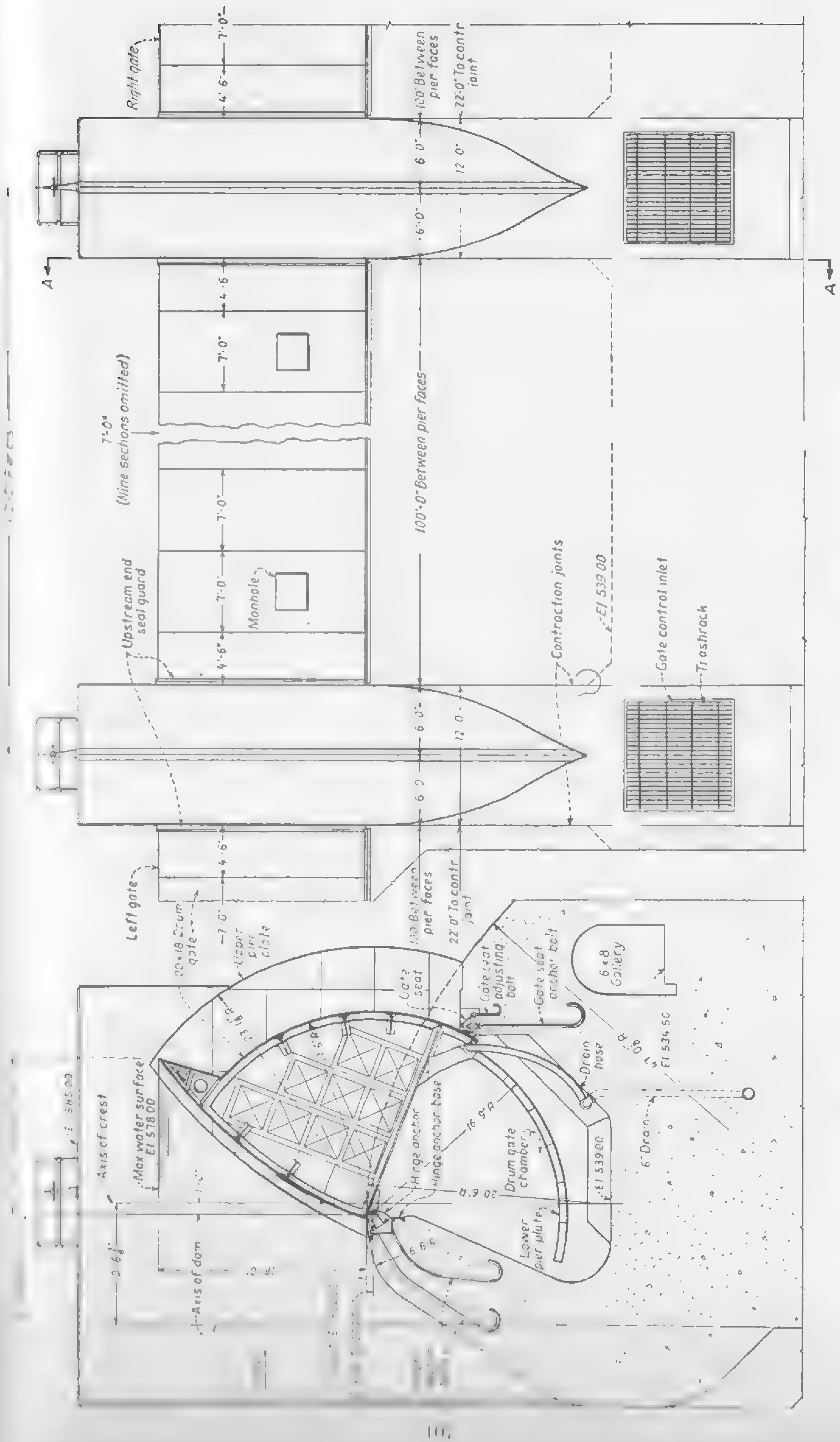


Fig. 6–1. Detail of automatic drum gate. (Bureau of Reclamation)

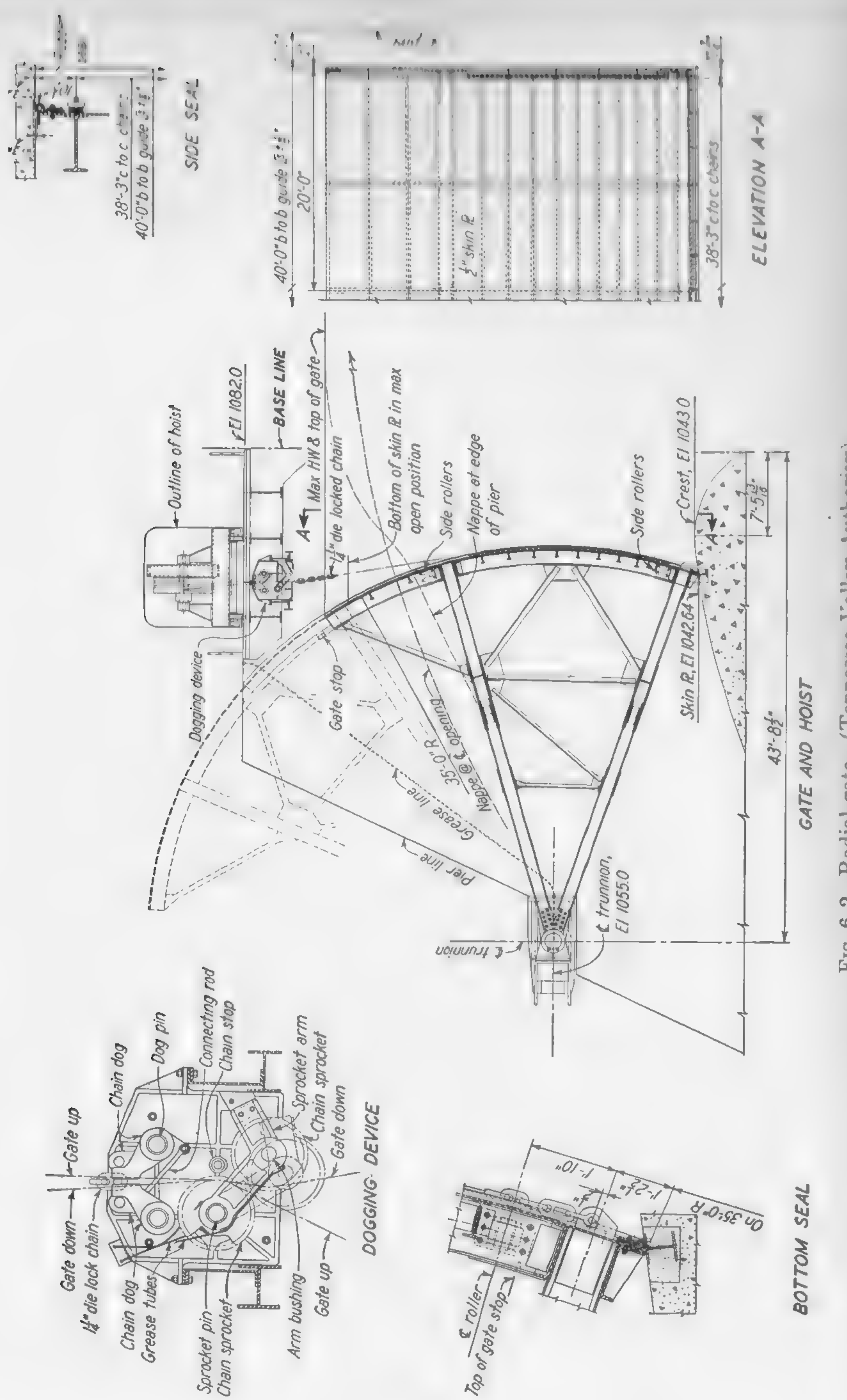


Fig. 6-2. Radial gate. (Tennessee Valley Authority)

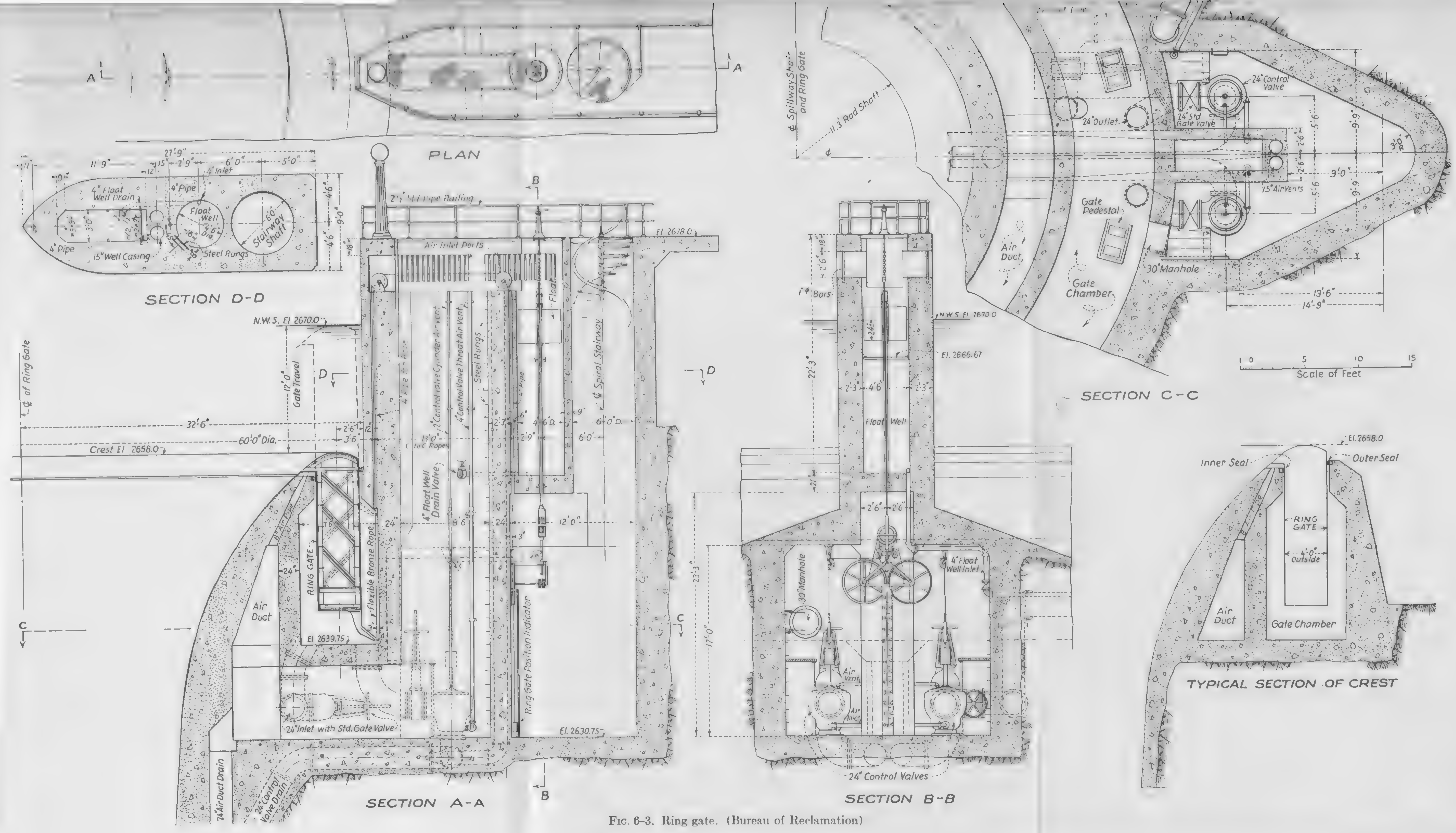


Fig. 6-3. Ring gate. (Bureau of Reclamation)

coefficient when the gate is in the fully open position. The force required to operate the gate is provided by the water pressure in the reservoir so that hoists and external power supply are eliminated. Drum gates are built up to 100 ft in breadth and from 12 to 28 ft in nominal height. The ratio of breadth to height ranges from 4 to 7, and the range of the arc of rotation is between 67 and 75 degrees.

Radial gates (Fig. 6-2) are built in the shape of a portion of a cylinder and rotate about a horizontal axis. The leaf consists of a face or skin plate supported by vertical side beams and horizontal beams. Braced steel arms extend from each side of the leaf to trunnion bearings anchored in concrete piers. Rubber seals at the sides and bottom control leakage. Wall plates and sill plates are attached to the sides and bottom with anchor bolts. Cables extending from the bottom of the gate to hoisting drums serve to lift the gate. Guide rollers operating along the wall plate serve to keep the alignment when the gate is being moved. The largest radial gate designed by the Bureau of Reclamation is 50 ft in width and 70 ft in height. This is considered a special type and is not a common installation.

Ring gates (Fig. 6-3) are an automatic type of gate which may be used to control "glory hole" spillways. They consist of a hollow annular drum sealed within a hydraulic chamber. The drum is raised by its own buoyancy in water introduced into the chamber from the reservoir, and it is lowered by release of water from the chamber. Its operation is similar to that provided for the hinged drum gate. The top of this drum is shaped as a spillway crest, so that when in a lowered position a high coefficient of spillway discharge is effective. The control may be automatic within certain ranges of reservoir levels, but hand controls are provided so that the gate may be held in a predetermined position.

Cylinder gates (Fig. 6-4) consist of a circular steel ring which moves up and down inside vertical guides. They may be designed for either external or internal water pressure. The cylinder is usually raised and lowered by three stems connected with screw-stem-type hoists. The stems are supported by guide bearings to prevent bending when compressive forces are exerted on the stems. They are commonly installed in intake towers.

Vertical lift gates include several types used for various purposes. They include: bulkhead gates, wheel- and roller-mounted gates, high-pressure control gates, and emergency gates.

Bulkhead gates usually consist of a vertical leaf which engages with the sealing element on sliding surfaces. They may be placed at the upstream end of a conduit to shut off flow or near the downstream end of the draft tube to permit dewatering. Horizontal cross beams carry

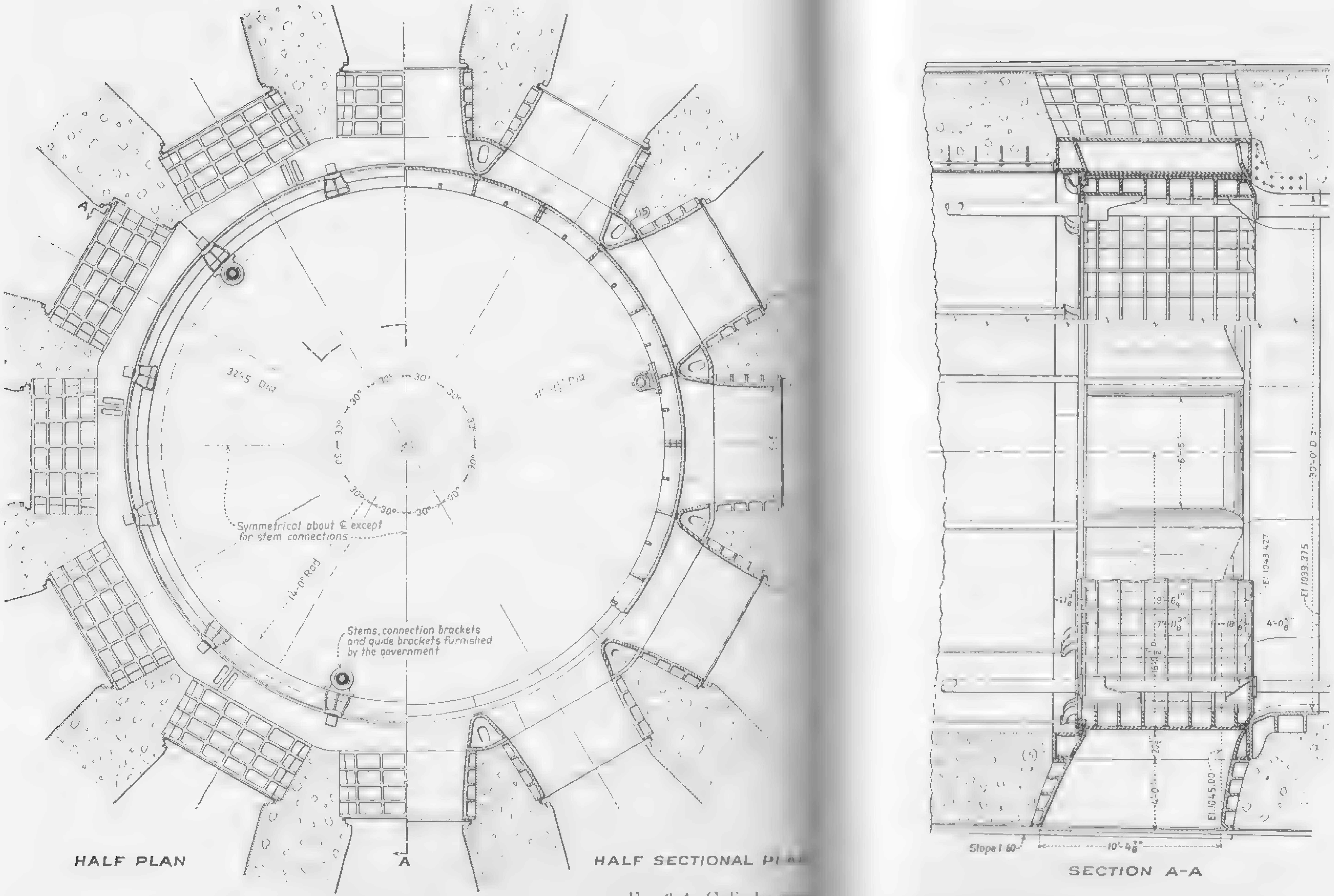


Fig. 6-4. Cylinder (Bureau of Reclamation)

FIG. 6-5. Traveling mechanical hoist. (Tennessee Valley Authority)

the water load to vertical girders at the sides. A skin plate carries the load to the cross beams. Rubber or brass seals bear against the seal slots. When not in use, the bulkhead gates may be stored in the upper portion of the gate guides. Hoisting or lowering may be done by cranes (Fig. 6-5). Where feasible, vertical lift gates may be divided into sections in order to minimize the size and capacity of handling or hoisting equipment, and to simplify problems of shipment and erection. Timber or steel stop logs are frequently substituted for bulkhead gates, and sometimes for crest or spillway controls.

Wheel-and-roller-mounted gates (Fig. 6-6) consist of a leaf composed of a face or skin plate fastened to horizontal beams or girders which are, in turn, supported by vertical girders. The horizontal girders of uniform cross section are variably spaced so that each takes an equal portion of the triangular water load. Due to the large size, the water load is so great that wheels or rollers must be supplied to reduce frictional resistance when the gate is raised or lowered. The term "coaster gate" is applied when continuous roller trains are mounted around the vertical girders. The roller trains operate on tracks in the face of the structure (Fig. 6-6). Coaster gates are used to control flow through penstock and outlet conduits. They are raised by hoists or cranes and lowered by their own weight.

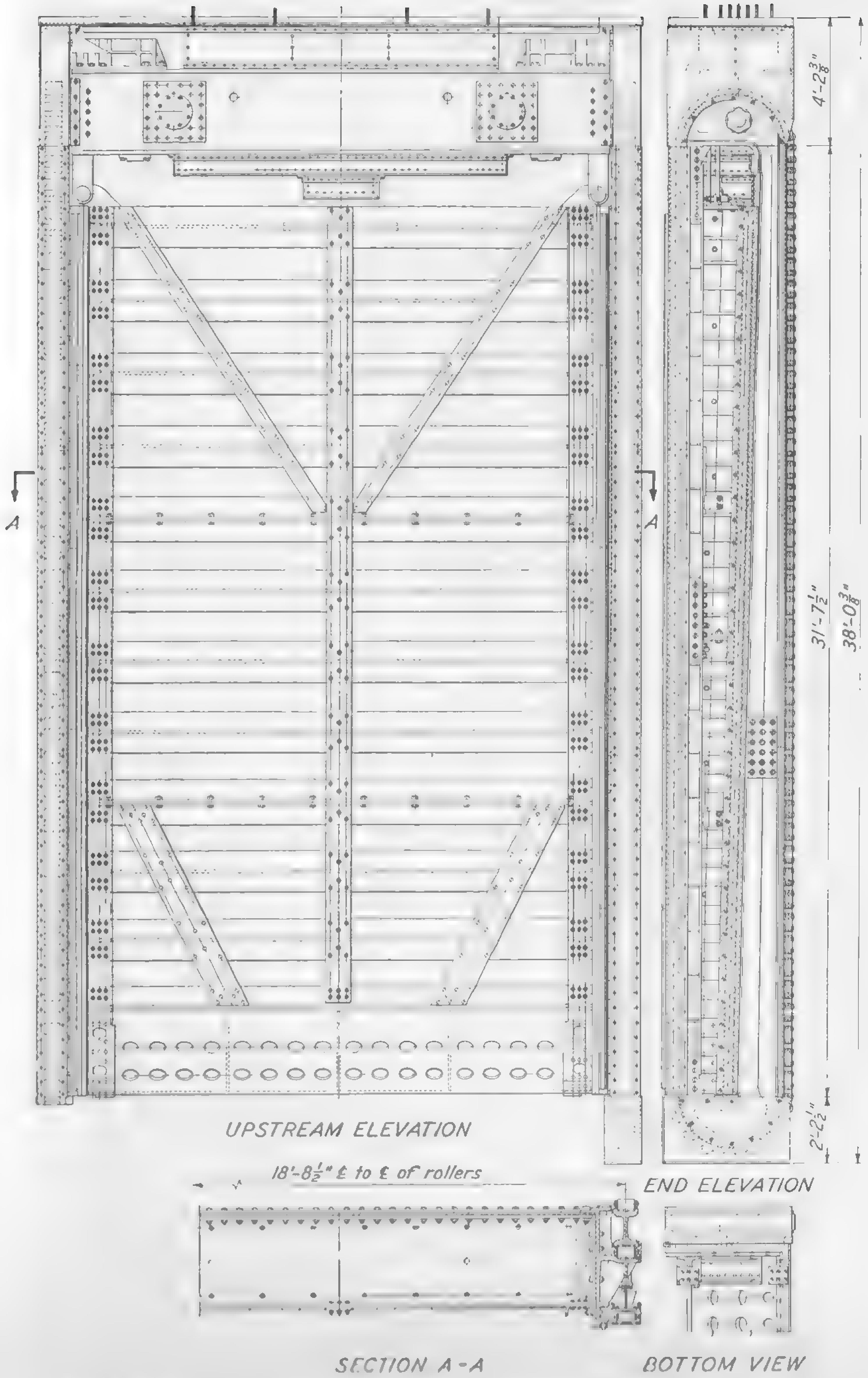


FIG. 6-6. Wheel-and-roller-mounted gate. (Tennessee Valley Authority)

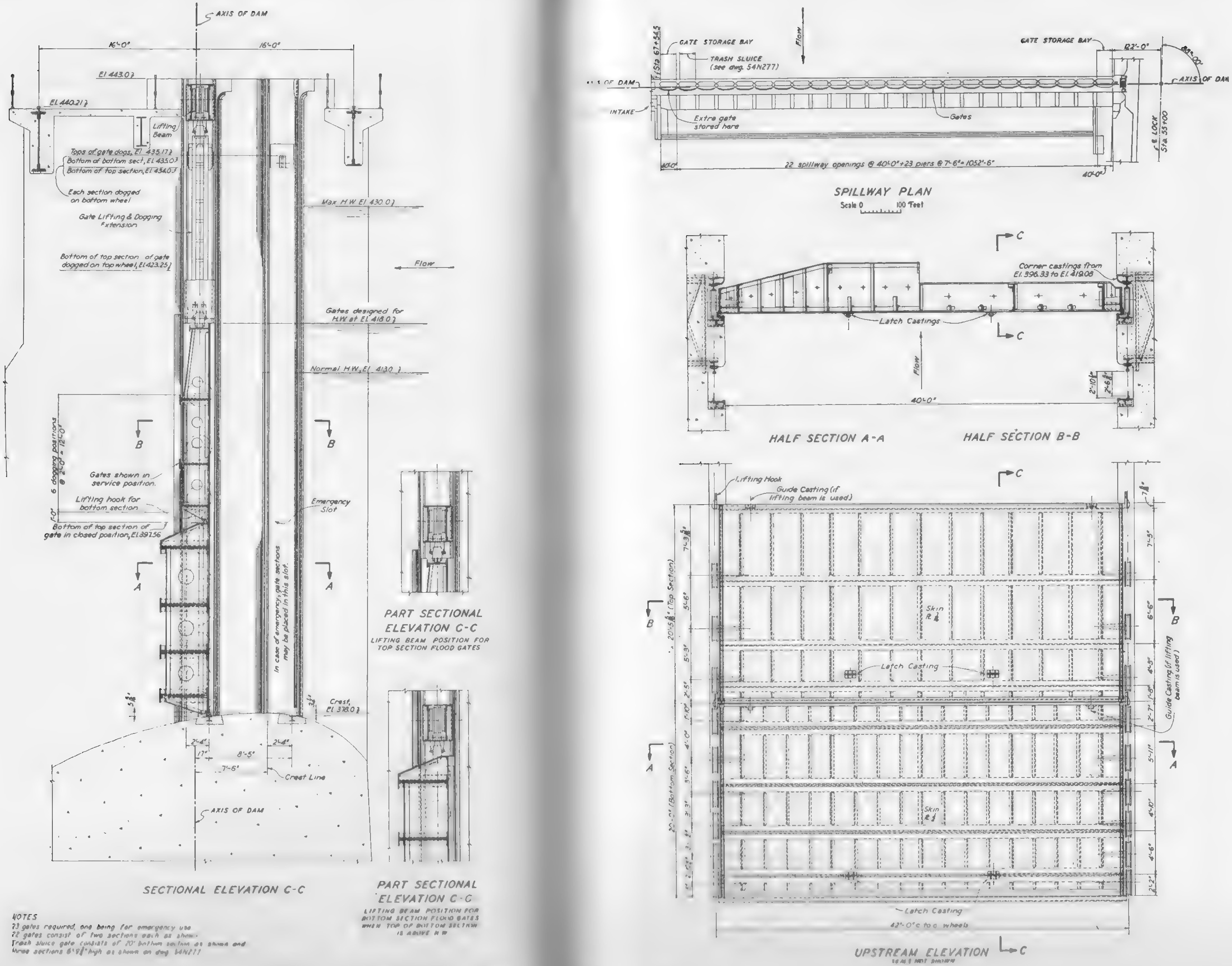


Fig. 6-7A. Two sections of a spillway (Tennessee Valley Authority)

Fixed-wheel gates are similar to coaster gates in general design and construction. Wheels are mounted between double vertical girders. On small gates, double-flanged wheels are used. On the larger gates, wheels without flanges are kept in line by spring-loaded guides at the sides of the gate and by disc springs on both sides of the wheel. Flexible, music-note seals provided with a thin brass quadrant vulcanized on the surface of the bulb are mounted on the skin plate. Fixed-wheel gates are used as controls for spillways and for tunnels, penstocks, and other conduits.

The Tennessee Valley Authority uses divided leaf gates (Figs. 6–7A

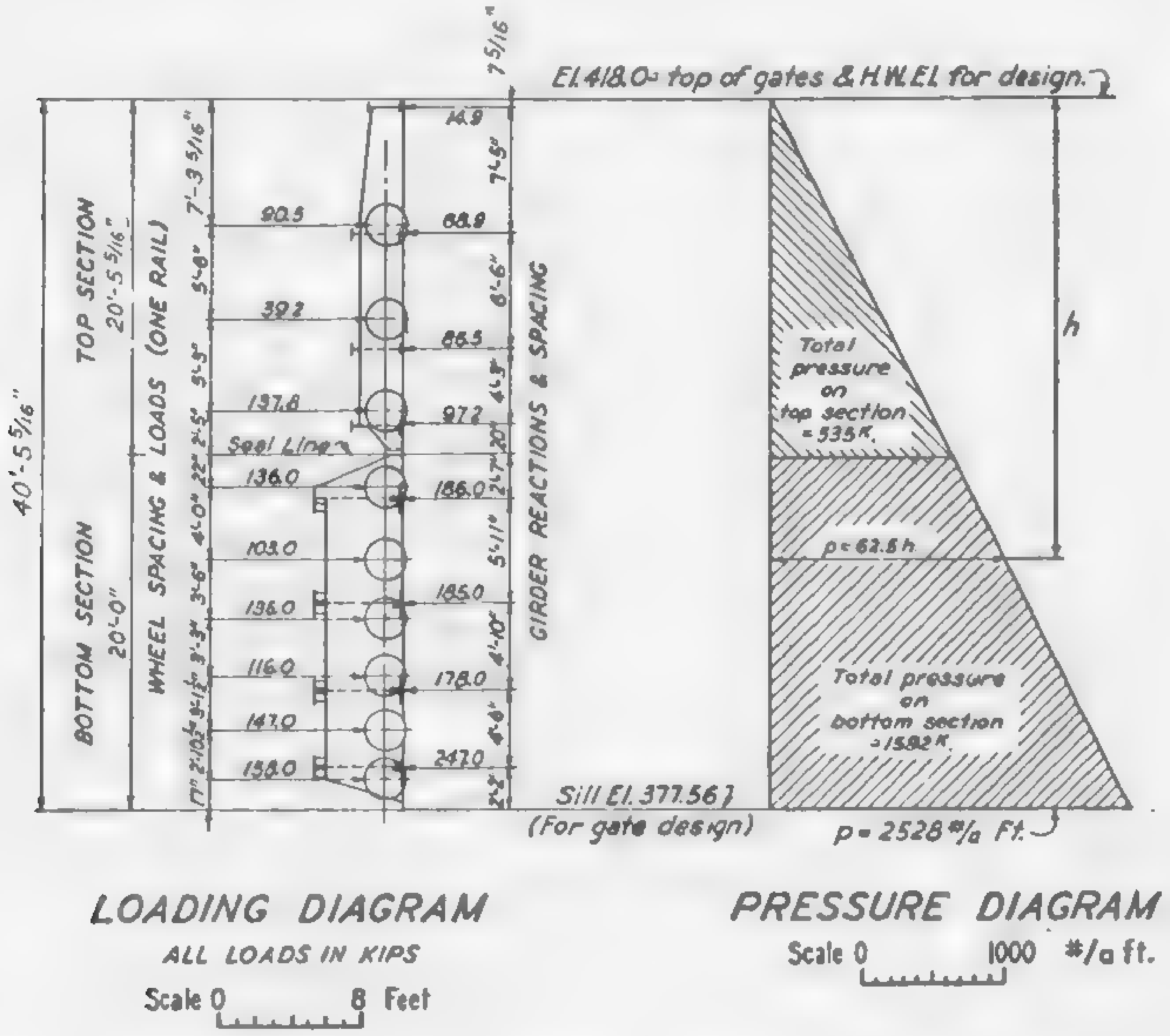


FIG. 6–7B. Loading diagram and pressure diagram of the spillway gate sections shown in Fig. 6–7A. (Tennessee Valley Authority)

and 6–7B) on several main-stream dams. The sections do not exceed 20 ft in height. The gates provide a clear spillway opening of 40 ft. At Guntersville and Chickamauga, the top section of one gate was divided into three sections so as to function as a trash gate. Rolled-steel double-flanged wheels 30 in. in diameter are provided for all gate sections. These wheels bear against downstream track rails and are guided by upstream guide rails installed in the spillway piers.

High-pressure control and emergency gates (Fig. 6–8) control the outlet works for dams. They are usually installed in tandem, so that the upstream gate can be used as an emergency gate and the downstream gate acts as the control. They are used for heads up to 250 ft. Gates of this type consist of a frame, bonnet, gate leaf, and hoist. One

essential feature of high-pressure gates is provision for supplying air to the downstream side of the outlet. This is necessary to prevent the development of negative pressures which would induce cavitation. The gate leaf may be of a square, rectangular, or circular shape. The modern trend is toward the circular shape. This avoids the necessity of transitions from square or rectangular to the usual circular form of most conduits. Various modifications of high-pressure gates include ring-follower gates, ring-seal gates, jet-flow gates, and others.

6–5. Valves. Valves are used to control flow in pipelines, conduits, and penstocks. The closing element operates and remains within the water passageway. The types of valves usually used in water power installations are: needle valves, Howell-Bunger valves, tube valves, hollow-jet valves, and butterfly valves. The selection of type will depend upon the head, impurities in the water, spray characteristics, and maintenance considerations. The valve types considered here are usually used for heads of 75 ft and greater. Water impurities such as sand or silt preclude the use of valves with close fitting parts that move upon each other. Spray from free-discharge valves may damage electrical installations. Avoidance of cavitation is of particular importance from the viewpoint of maintenance. The ease and probable frequency of repairs are of special importance in considering the cost of operation and reliability of service. The inital cost of valves varies approximately as the weights, and the weights vary approximately as the cube of the inlet diameter.

Needle valves are devices used to control reservoir outlets which must operate under high heads. The needle valve consists of a pointed piston or needle, which slides in an internal cylinder (Fig. 6–9). The cylinder is installed in a truncated casting and is held in place by ribs connecting it with the outside shell. The needle is moved by water pressure from the outlet conduit which acts upon the interior chambers of the valve. The most modern type is known as the internal differential needle valve. These valves range in size from 15 to 102 in. in inlet diameter, and may be designed to operate under heads from 75 to 600 ft. The discharge coefficient C in the formula $Q = CA\sqrt{2gh}$ has a value of 0.58. A is the area of the inlet in square feet, and h is the effective head on the center line of the valve one diameter upstream from the flange.

The *Howell-Bunger valve* (Fig. 6–10) is a patented type of advanced design which controls water under either high or low heads. The valve is composed of three parts: body, gate, and operating mechanism. The body consists of a steel plate or cast steel cylinder, an inverted cone-shaped head, and internal ribs. The radial ribs extend beyond the downstream end of the cylindrical shell through the valve ports to the

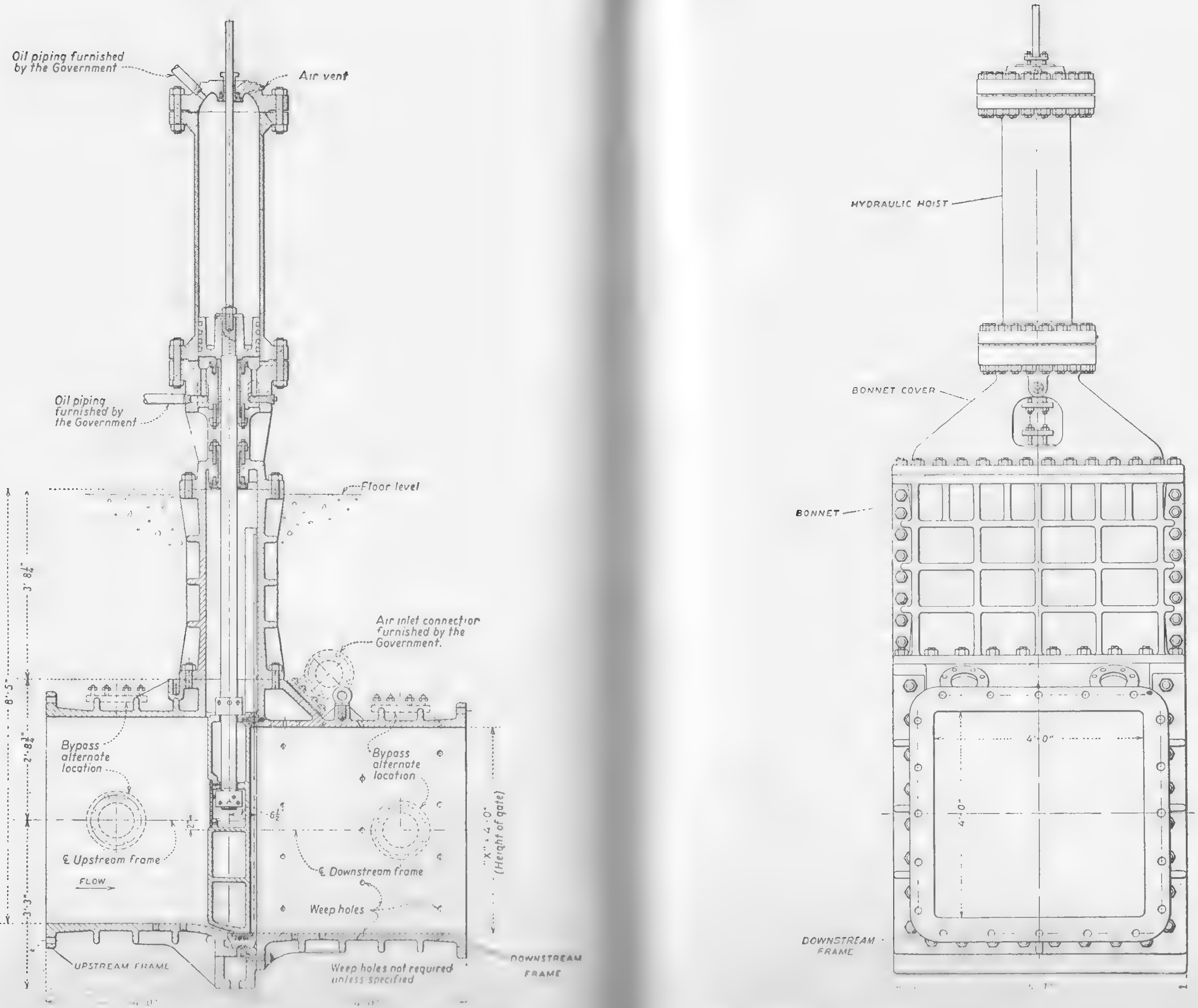


FIG. 6.8. High-pressure control gate. (Bureau of Reclamation)

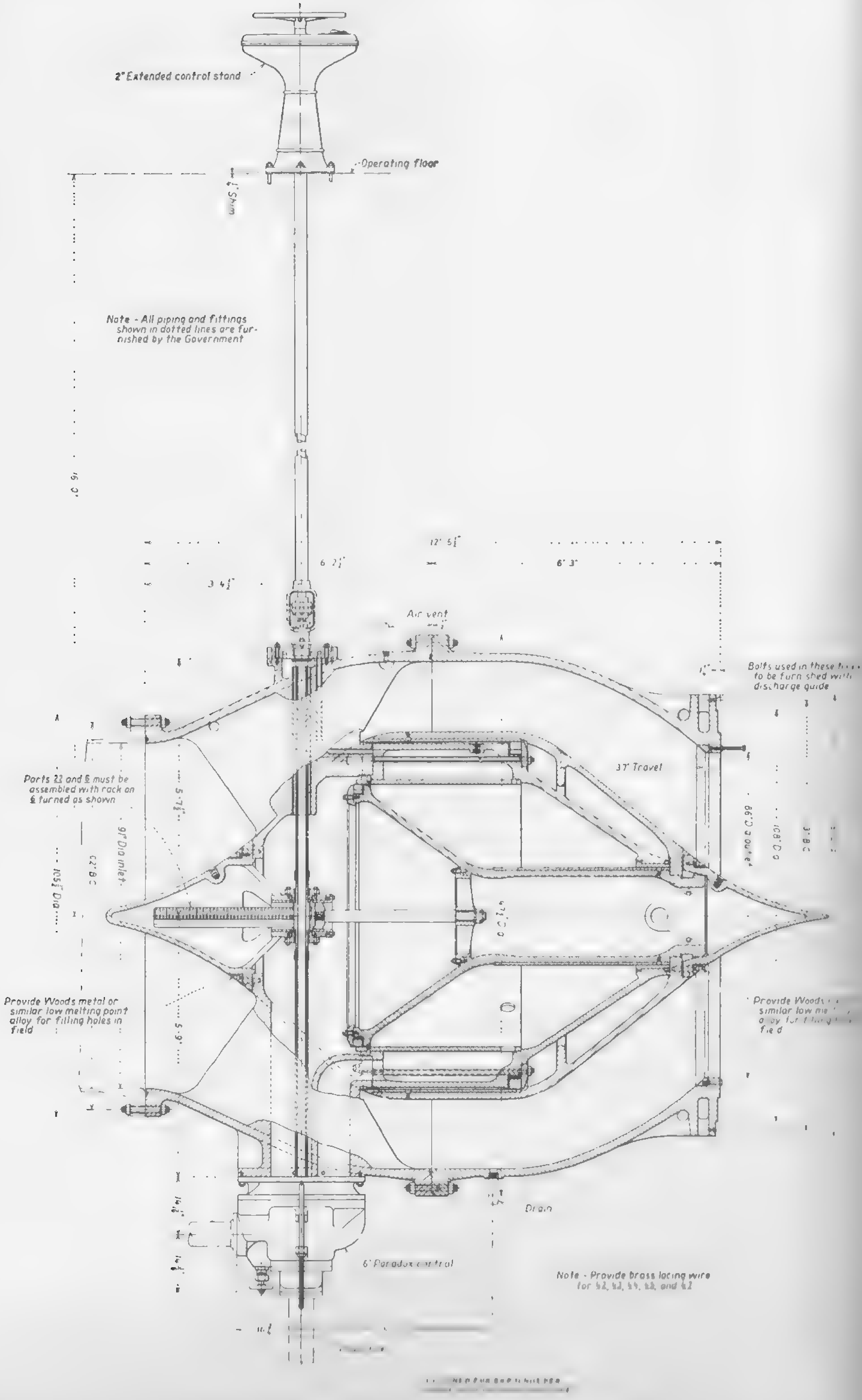


Fig. 6-9. Needle valve. (Bureau of Reclamation)

Fig. 6-10. Howell-Bunger valve. (S. Morgan Smith Co.)

cone-shaped head. The cylinder gate slides on a polished brass sleeve fitted over the cylindrical body. The inside of the cylinder gate is machined to a sliding fit over the brass sleeve. The downstream end of the gate has a stainless steel seal ring which seats against a beveled stainless steel seat ring. Graphite packing provides a seal between the gate and the body. The operating mechanism is placed on the outside of the body and consists of two bronze nuts, two bronze screw stems with steel pipe covers, two worm gear reducers, and one mitre gear box. The valves may be manually operated with a handwheel or motor-operated with a high-torque motor, equipped with ball bearings.

The Howell-Bunger valve has several advantages, among which are low cost, easy operation, energy-dissipating characteristics, and high discharge coefficient. When operating as a free discharge valve into the air, it breaks up the water stream into a large hollow expand-

ing jet. A recommended formula for discharge when fully open is $Q = 0.85A\sqrt{2gh}$ or $Q = 5.354D^2\sqrt{h}$, in which Q is the discharge in cubic feet per second, A is the area in square feet, h is the head in feet on the center of the valve, and D is the inside diameter of the body in feet. The valves are built in sizes ranging in diameter from 8 in. to 9 ft, and for heads ranging up to 420 ft for the larger sizes and up to 900 ft for the smaller sizes. The length of an 18-in. valve of this type is 4 ft 4 in., and the length of a 9-ft valve is 13 ft 4 in.

Tube valves (Fig. 6–11) were developed from the internal-differential needle valve. The water passages are similar but the downstream end of the needle is omitted. A hollow cylinder similar to that of a cylinder gate constitutes the moving part of the valve. Tube valves may be used internally in a conduit or at the discharge end. The internal type has a long body with a 30-degree nozzle, whereas the free discharge type has a short body and a 45-degree nozzle. Air inlets must be provided to aerate the downstream jet to protect against cavitation. The coefficient of discharge C in the formula $Q = CA\sqrt{2gh}$ is about 0.52 for the short-body free-discharge type and 0.72 for the long-body internal type. These values apply only to fully open conditions. The free-discharge type should be operated between 30 and 100 per cent of the opening range, since the jet becomes unstable at openings less than 30 per cent.

Hollow-jet valves are essentially needle valves in which, as in the tube valve, the downstream needle is omitted. Closure is effected by moving the upstream needle counter to the direction of flow. They are used only at the downstream end of a conduit. Their water dispersion characteristics are similar to those of the Howell-Bunger valves. The coefficient of discharge is approximately 0.70 for full valve opening. Sizes of jet valves range from 16 to 120 in.

Butterfly valves (Fig. 6–12) have a circular or conical body with a circular leaf mounted on a vertical or horizontal shaft which has a bearing at each end. An external operating mechanism rotates the leaf through a 90-degree arc from a closed to an open position. Leakage around the leaf is controlled by an adjustable sealing device. The body and the leaf must be shaped in such a way that abrupt changes in velocity will be reduced to a minimum. Electric motors may be used to operate small valves, and hydraulic cylinders are used for the large sizes. Butterfly valves are frequently used in penstocks, upstream from the turbine case, as service or emergency control gates. They are also used as shutoff or regulating valves in outlets, but they are not entirely satisfactory for operation at partial opening due to possible cavitation effects. They are built in sizes up to 27 ft in diameter.

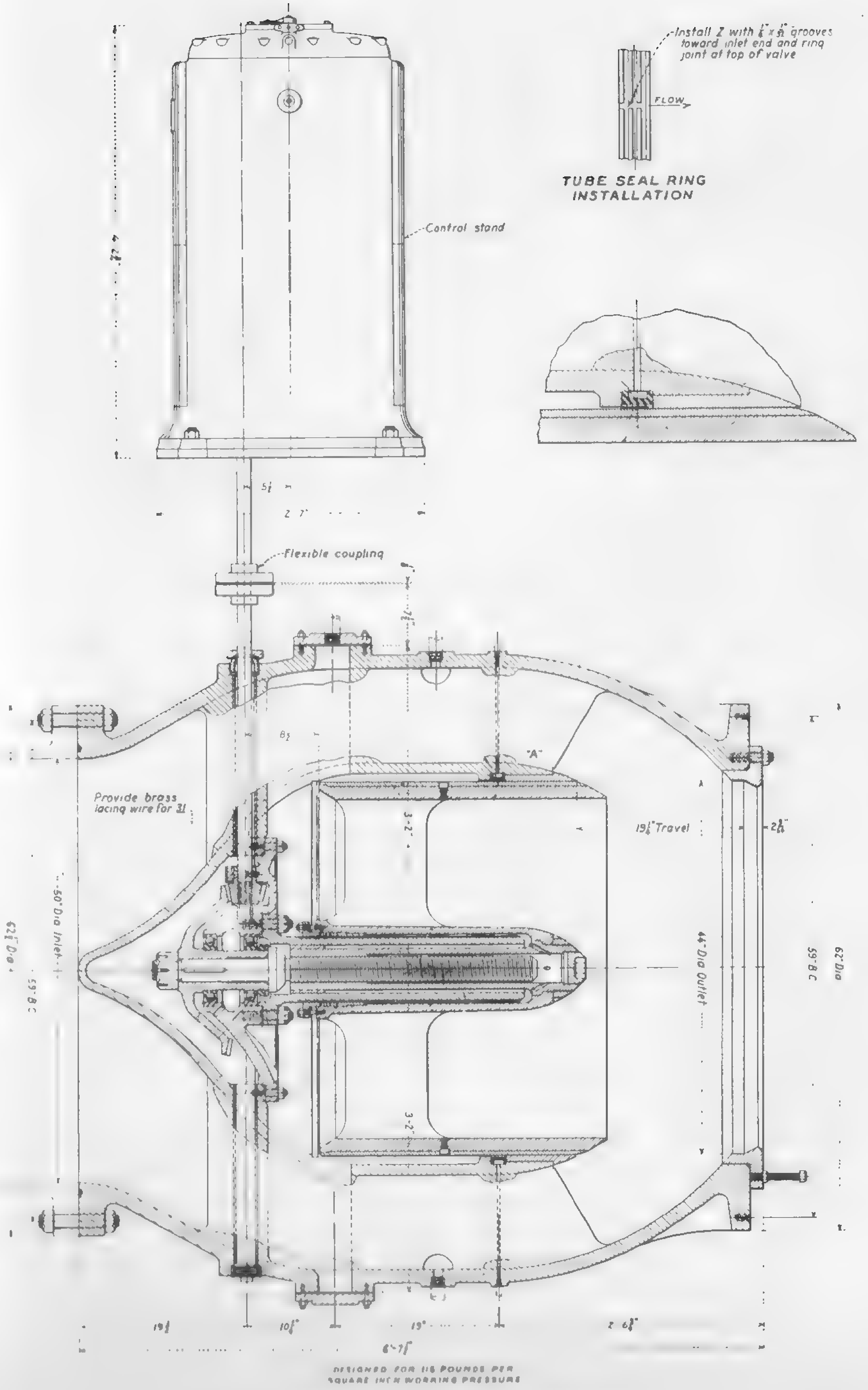


FIG. 6-11. Tube valve. (Bureau of Reclamation)

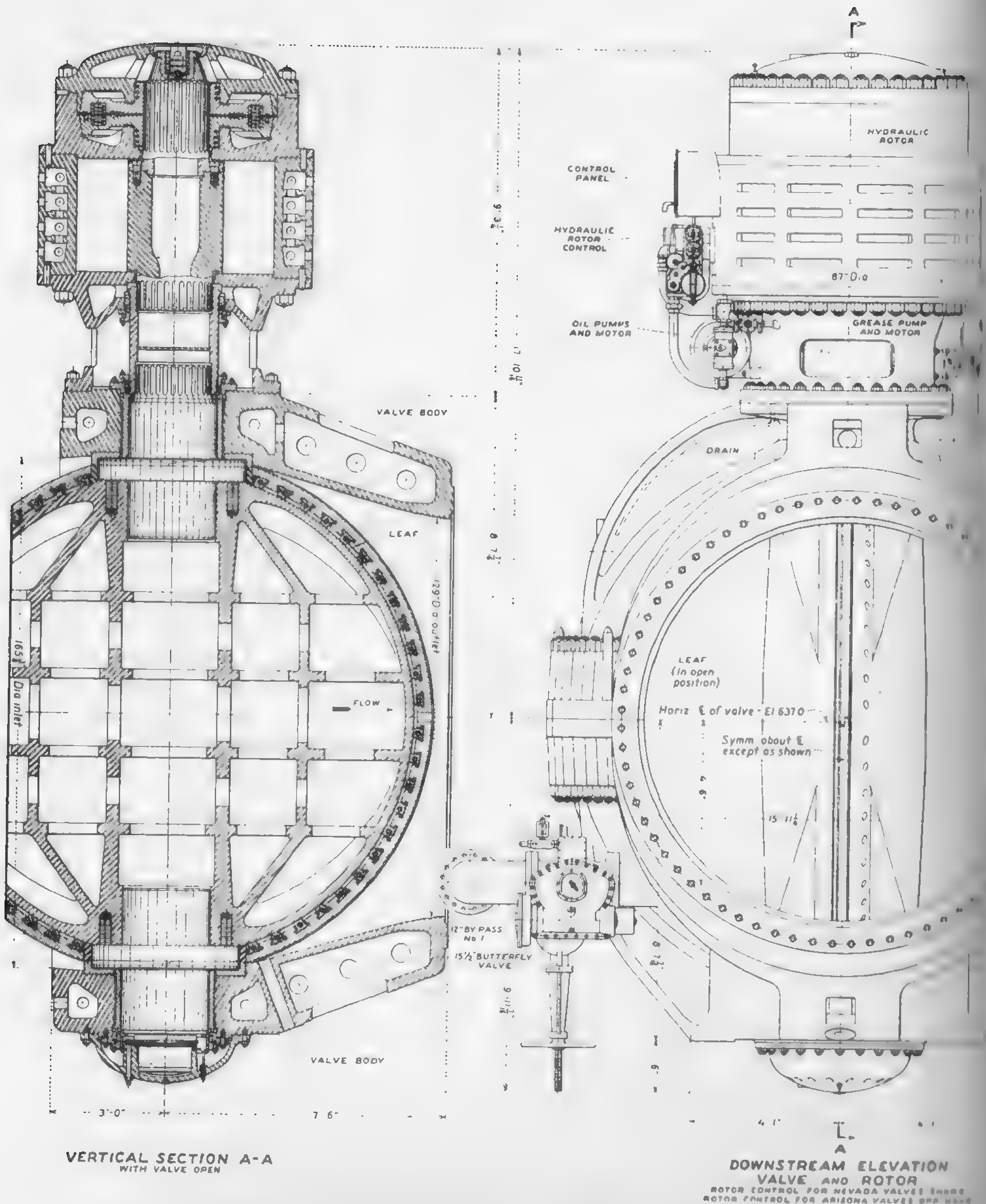


FIG. 6-12. A 168-inch butterfly valve. (Bureau of Reclamation)

6-6. Waterways. Waterways include all channels which serve as conveyances for water from the source in a river or reservoir to the tailrace of a power plant. In this section the discussion will include canals, penstocks, pipelines, and tunnels, but will not treat short channels leading to the entrance of a flume or scroll case, and all passages to, around, and from the turbine.

Canals are hydraulic channels excavated in earth or rock which operate under the laws of open channel flow. The capacity in cubic feet per second is dependent upon the cross section and velocity. The velocity is dependent upon hydraulic radius, friction coefficient, and slope. The design of a power canal must take into account the economic factor of the relation between the annual cost of the capital cost of construction and maintenance against the loss of head and, consequently, against the loss of power potential at the turbine. Detailed design practices are not included here for it is assumed that the advanced civil engineering student is familiar with canal design. Suffice it to say that the reduction of head loss and annual maintenance cost afforded by concrete-lined power canals usually warrants the adoption of a lining. An example of a power canal is the New Beauharnois Canal, which carries water from Lake St. Francis around the rapids of the St. Lawrence River to the power plant on Lake St. Louis.

Flumes are a special type of canal, and their design is governed by the same hydraulic characteristics. They are used to carry water over topographic depressions and along sidehills. Flumes are built of concrete, steel, and wood. The design of intake and outlet transitions are important to the saving of head, when the flume cross section is smaller than that of the canal it serves.

Penstocks and pipelines (Fig. 6-13) are conduits used to convey water to hydraulic turbines. They are usually built of steel, either riveted or welded. Some penstocks and pipelines are built of wood in areas where suitable materials are available and at times when steel supplies are scarce. Concrete is also used as a closed conduit under relatively low head. Pipelines are used as siphons or sag pipes to carry water over topographic depressions, such as canyons and gullies, and across streams.

In such conduits the water flows under pressure and the hydraulic considerations follow the laws of pressure flow. Each proposed installation requires a study of economic and hydraulic factors which determine the required dimensions of the structure. The relation between head loss and power potential is of great importance.

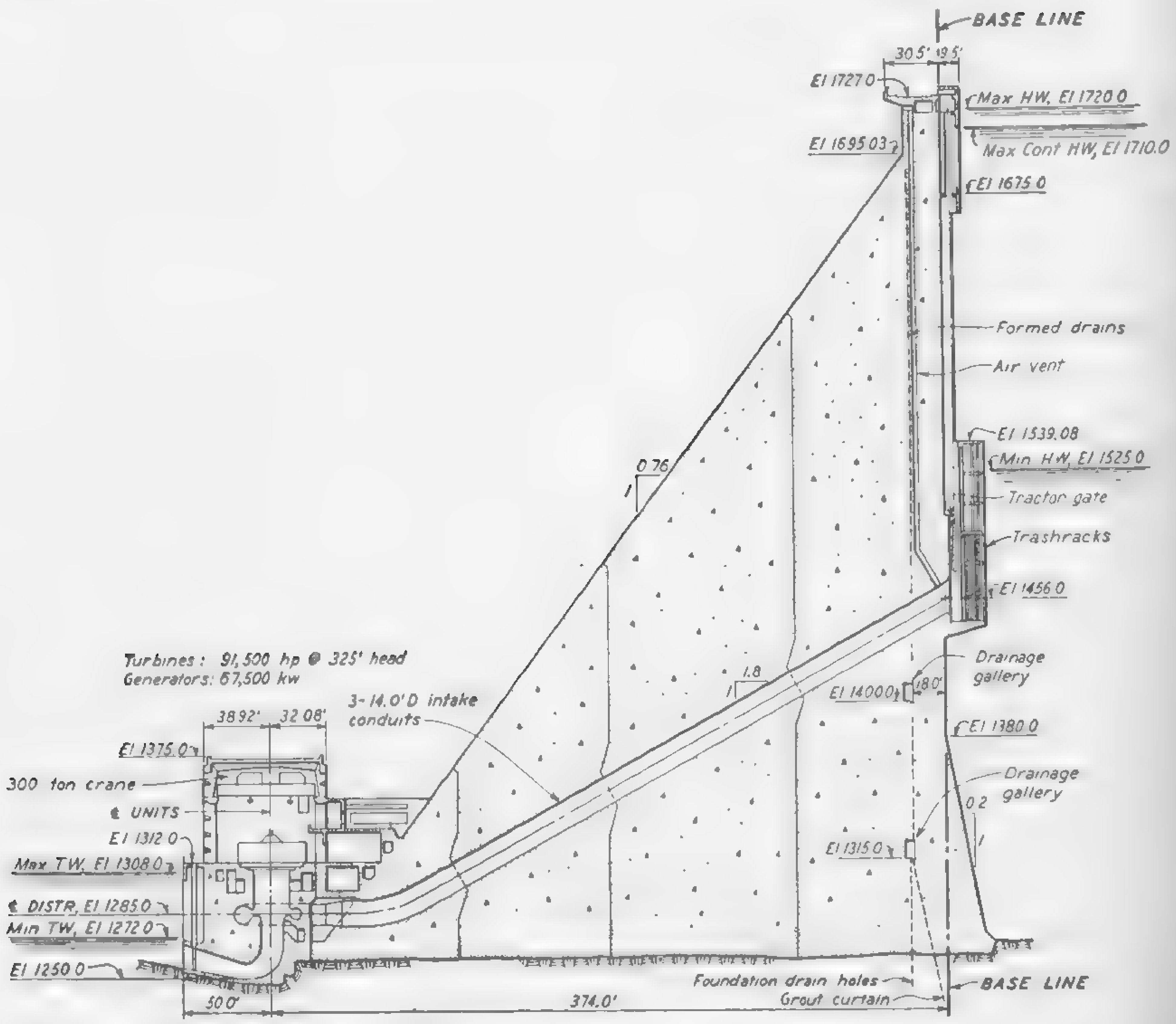


FIG. 6-13. Installation of steel penstock. Note the trashracks and control gate at inlet. (Tennessee Valley Authority)

6-7. Head Losses, Steel Pipe. The head loss due to pipe friction in straight steel pipe may be computed from the Scobey formula:

$$H_f = K_s \frac{V^{1.9}}{D^{1.1}}$$

where H_f = head loss in feet per 1000 ft of pipe
K_s = an experimental coefficient
V = velocity of flow in feet per second
D = diameter of pipe in feet

The value of K_s varies from 0.32 for new continuous-interior pipe (welded) to 0.52 for butt-strap plate metal greater than ½ in. in thickness. The effect of aging of the pipe is computed by multiplying K_s by $e^{0.015t}$, where e is the base of natural logarithms and t is the age of the installed pipe in years of service.

The head losses in bends may be computed by the Hinds formula: *

$$h_b = C \cdot \sqrt{\frac{\Delta}{90}}\left(\frac{V^2}{2g}\right)$$

where h_b = bend loss in feet
C = an experimental coefficient
Δ = deflection angle of bend in degrees
V = velocity of flow in feet per second

The losses vary also as the ratio of the radius of the bend R and the diameter of the pipe D, both in feet. In the Hinds formula, $C = 0.25$ when R/D is equal to or greater than 2. R/D ratios of 3 to 4 are commonly used for pipelines.

Losses through trashracks may be computed as 0.1, 0.3, and 0.5 ft, respectively, for velocities of 1.0, 1.5, and 2.0 fps.

Entrance losses depend upon the shape of the intake opening. The circular bellmouth opening, proportioned as shown in Fig. 6-14, is considered to be the most efficient shape. Losses through such an opening will be between 0.05 and 0.10 of the velocity head. Estimated losses through square bellmouth entrances are estimated at 0.2 of the velocity head. In entrances not provided with a bellmouth, the loss of head due to entrance may range from 0.5 to 0.85 of velocity head. It should be remembered in connection with entrance losses that the given values are in addition to the head drop necessary to produce the velocity of flow in the conduit.

Head losses resulting from divided flow in Y branches of penstocks depend upon the following: (1) ratio of the diameter of the branch to the diameter of the penstock, (2) the angle Δ between the alignment of the branch and that of the penstock, and (3) the ratio between the rate of discharge Qa cfs in the branch to the rate of discharge Q in the penstock upstream from the branch. Right-angle branches and cylindrical outlets induce excessive losses of head and are susceptible of cavitation troubles. A conical outlet with sidewalls making an angle of from 6 to 8 degrees with the axis of the branch reduces losses up to one third of those with cylindrical connections.

6-8. Design of Steel Penstocks and Pipelines. The thickness of the pipe shell, t, is determined from the formula:

$$t = \frac{pR}{f_t e}$$

* Bureau of Reclamation, Tech. Memo No. 10

where p = internal pressure in psi
R = radius of the pipe in inches
f_t = allowable unit stress in psi
e = joint efficiency.

The value of p should include all water hammer effects. The allowable unit stress (f_t) should be one half the yield point for open or insufficiently embedded pipe. If the embedment provides a cover equal to a minimum of one half of the static head, a design stress of two thirds of the yield point may be used. The efficiency e may be taken as unity for welded pipe, if the welds are to be completely radiographed. Otherwise, a value of $e = 0.90$ should be used. If riveted joints are to be used, the efficiency of plate or the efficiency of rivets, whichever is the smaller value, should be used. The pipe thickness should be such that the allowable unit stress times the efficiency is not exceeded for all conditions of hoop tension and circumferential and longitudinal stresses. Such additional stresses may develop if the pipe is freely supported at several points along its length or if it acts as a cantilever between the support and expansion joint. Stresses may also be caused by frictional

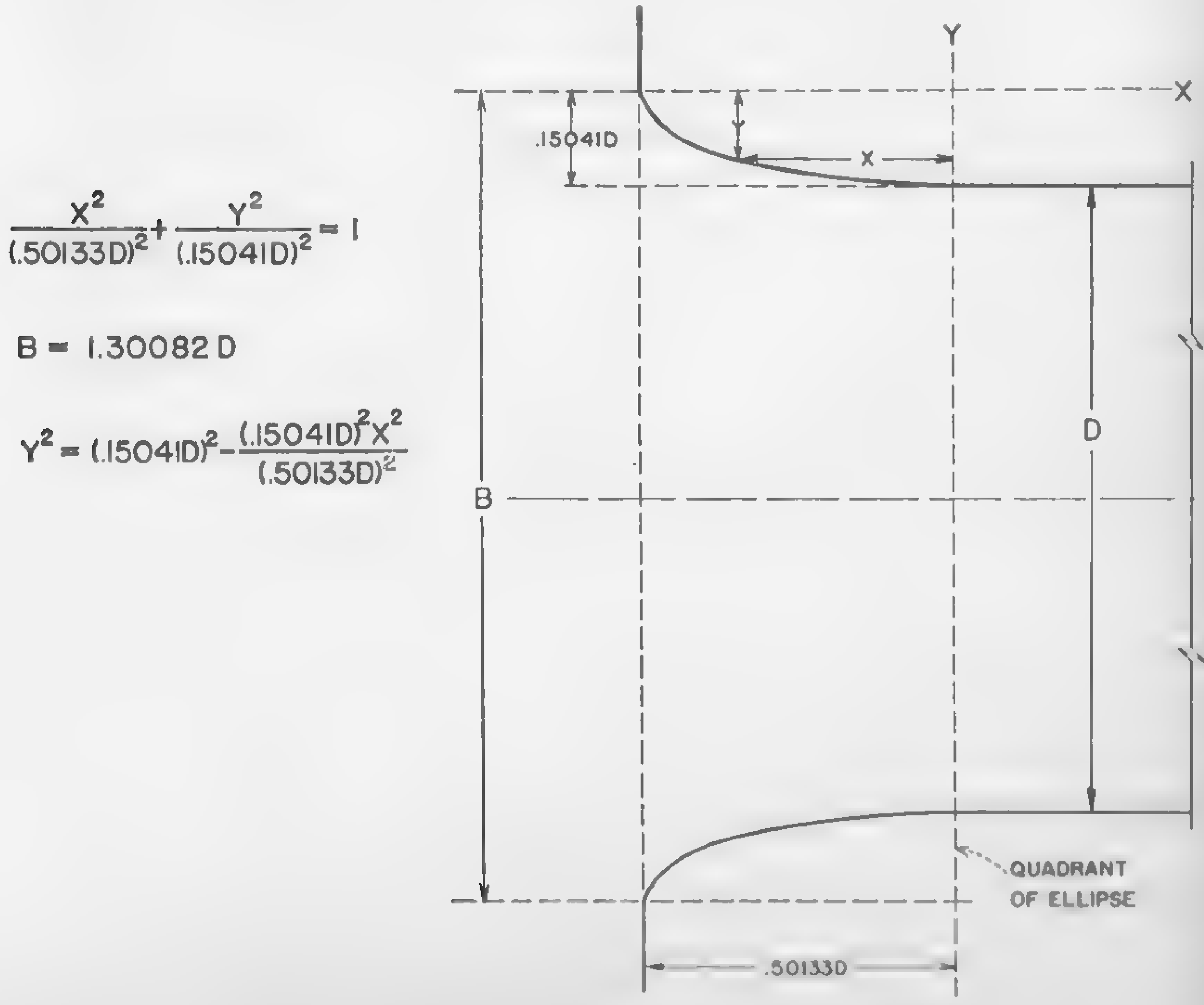


Fig. 6-11. Detail of ideal bellmouth entrance.

resistance of the supports when temperature changes cause longitudinal movement.

Minimum thickness for steel penstocks may be computed by

$$t\ (\text{min}) = \frac{R+10}{200}$$

and for outlet pipes carrying high velocity,

$$t\ (\text{min}) = \frac{R+16}{130}$$

One sixteenth of an inch is sometimes arbitrarily added to the pipe thickness to care for expected corrosion.

Where conduits act as beams between pier supports, the maximum equivalent unit stress should not exceed the critical buckling stress S.

$$S = C\,(0.6E)\frac{t}{R}$$

where S = critical buckling stress in psi
C = constant varying from 0.20 for a shell thickness of 0.03 in., to 0.26 for a shell thickness of 0.25 in.
E = modulus of elasticity in psi
t = mean thickness of shell
R = mean radius of conduit shell in inches

The economical diameter of a steel penstock may be determined from a study of the annual charges on the cost of the installed pipe versus the value of the power lost through hydraulic losses. Experience indicates that the economical velocities will vary between 10 and 20 fps. Preliminary determination of the economical velocity V and economical diameter D in a penstock for heads over 100 ft may be made through the use of empirical formulas. P. J. Bier suggests the following:

$$V = 0.125\sqrt{2gh}$$

or

$$D = \left(\frac{P}{H}\right)^{0.466}$$

where V = economical velocity
$\sqrt{2gh}$ = theoretical spouting velocity
P = rated horsepower
H = rated head of a Francis turbine

The velocity formula fits actual installations better than the diameter formula. In some instances, particularly for very high heads, the pen-

stock diameter is reduced toward the power plant, because the saving in first cost more than offsets the reduction in power potential through the higher friction losses.

The design of stiffener rings and support rings for steel pipe and the design of pipe at branch outlets and wyes belong in a specialized field and are not treated here.*

6–9. Design of Penstock Piers and Anchors. Unburied penstocks are usually supported on piers spaced from 20 to 60 ft apart. Anchors are placed at bends in order to resist forces which are apt to displace the pipe. Piers and anchors act as a foundation for the pipe. The characteristics of the soil or rock on which they rest are important considerations for their design. (See Fig. 6–15.)

Piers support the dead weight of pipe and water and resist longitudinal forces caused by movement of the pipe due to temperature changes. These forces are transmitted to the pier through friction when the pipe rests either directly upon the concrete or on bearing plates. The magnitude of the longitudinal forces may be reduced by the use of lubricated plates, graphited service sheets, or rocker or roller supports. The dimensions of the pier are determined from an analysis of all forces acting upon it. The usual requirements of stability are observed in that the resultant of all forces will intersect the base within its middle third.

Anchors are similarly designed except that the following forces must be considered:

1. Hydrostatic force acting along axis of pipe on each side of bend

$$= wAH$$

where w = weight of water per cubic foot = 62.5 lb
A = cross-sectional area of pipe in square feet at anchor
H = maximum head at any point, including water hammer, in feet

2. Dynamic force acting against outside of bend

$$= \frac{qwv}{g}$$

where q = flow in cubic feet per second
v = velocity in feet per second
g = acceleration due to gravity in feet per second = 32.16

* The reader is referred to Herman Schorer, "Design of Large Pipe Lines," *Trans. A.S.C.E.*, XCVIII (1933), 101-91; and P. J. Bier, *Welded Steel Penstocks*, Bureau of Reclamation, Engineering Monograph No. 3, July, 1949 (Washington, D. C.: Government Printing Office, 1949).

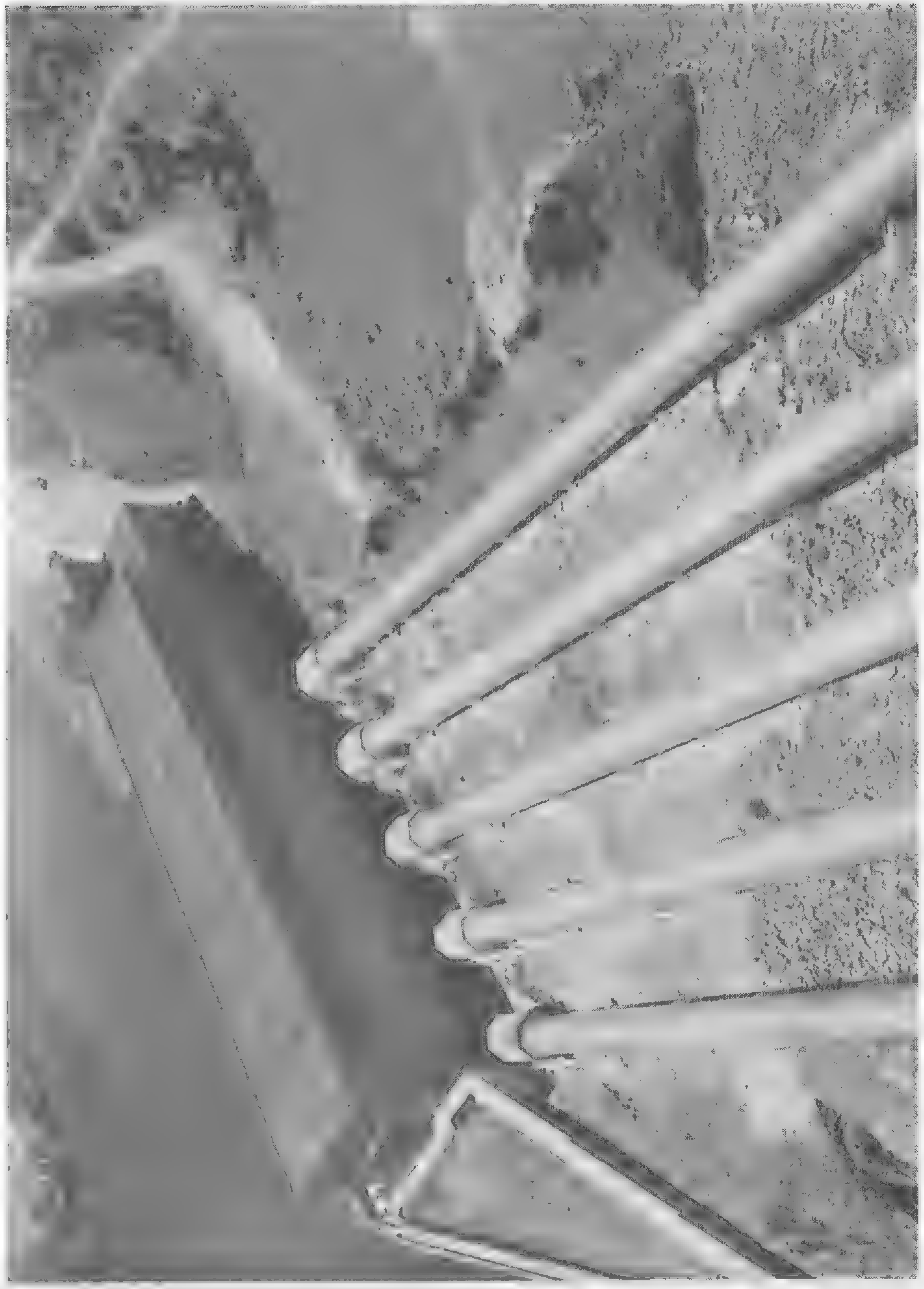


Fig. 6–15. Penstocks of the Shasta Dam power plant, showing pipe supports and anchors. (Bureau of Reclamation)

3. Force due to dead weight of pipe from anchor uphill to expansion joint, tending to slide downhill over piers

$$= P \sin x$$

where P = dead weight of pipe from anchor uphill to expansion joint in pounds
x = slope angle above anchor

4. Force due to dead weight of pipe from anchor downhill to expansion joint, tending to slide downhill over piers

$$= P' \sin y$$

where P' = dead weight of pipe downhill from anchor to expansion joint in pounds
y = slope angle below anchor

5. Sliding friction of pipe on piers due to expansion or contraction uphill from anchor

$$= f \cos x \left(P + W - \frac{p}{2}\right)$$

where f = coefficient of friction of pipe on piers
W = weight of water in pipe
p = weight of pipe and contained water, from anchor to adjacent uphill pier, in pounds

6. Sliding friction of pipe on piers due to expansion or contraction downhill from anchor

$$= f \cos y \left(P' + W - \frac{p'}{2}\right)$$

where P' = dead weight of pipe downhill from anchor to expansion joint in pounds
p' = weight of pipe and contained water, from anchor to adjacent downhill pier, in pounds

7. Sliding friction of uphill expansion joint

$$= \frac{f'\pi(d + 2t)}{12}$$

where f' = friction of expansion joint per linear foot of circumference = approximately 500 lb
d = inside diameter of pipe in inches
t = thickness of pipe shell in inches

8. Sliding friction of downhill expansion joint

$$= \frac{f'\pi(d + 2t)}{12}$$

9. Hydrostatic pressure on exposed end of pipe in uphill expansion joint

$$= waH$$

where a = cross-sectional area of pipe shell at uphill expansion joint in square feet

10. Hydrostatic pressure on exposed end of pipe in downhill expansion joint

$$= wa'H$$

where a' = cross-sectional area of pipe shell at downhill expansion joint in square feet

11. Longitudinal force due to reducer above anchor

$$= wH(A' - A)$$

where A' = cross-sectional area of pipe above upper reducer in square feet

12. Longitudinal force due to reducer below anchor

$$= wH(A - A'')$$

where A'' = cross-sectional area of pipe below lower reducer in square feet

Figure 6–16 shows, in profile, the direction of the various forces for expanding and contracting conditions; each force must be resolved into its vertical and horizontal components. In the above analysis the restraint exerted by the pipe itself against overturning of the anchor is not considered. It must also be remembered that the weight of the anchor must be included in stability studies. In some cases, the rigid application of the forces 1 to 12 results in structures which are inordinately large. The exercise of good engineering judgment, based upon successful installations, is important to the development of a reasonable, yet safe, design.

6–10. Air Inlets. Air inlets are provided in the upstream ends of penstocks and outlet pipes (1) to avoid the development of negative

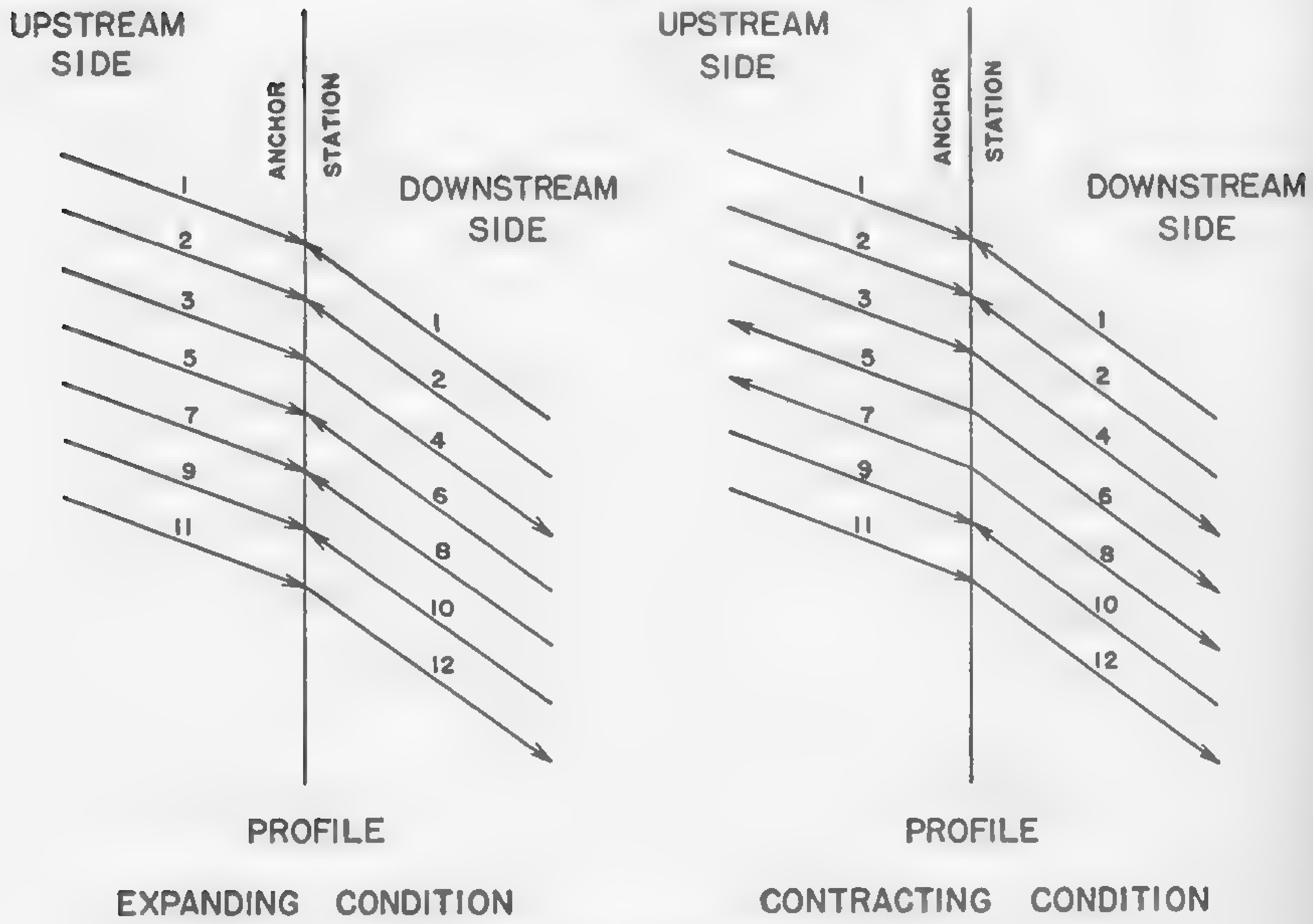


Fig. 6–16. Forces acting on anchors.

pressure inside the pipe when the conduits are being drained, and (2) to release air during filling. Air inlets should also be provided on summits. Enger and Seely developed the following formula for flow of air:

$$Q = 348cF\sqrt{P}$$

where Q = flow of air in cfs
c = coefficient of discharge through air inlet (approximately 0.7)
F = area of air inlet in square feet
P = safe differential in pressure between the inside and outside of the pipe in psi

Other accessories for pipelines and penstocks include manholes, drain and filling connections, and piezometer connections.

6–11. Tunnels. Tunnels are conduits excavated through earth or rock. The cross-sectional area is determined by economic considerations. For example, the Tennessee Valley Authority considers the power value to be from 2 to 3 mills per kwh; and the coefficient of friction in the Manning formula is assumed to be 0.012 for lined tunnels and 0.035 for unlined tunnels. Tunnels may carry water under open-

channel flow conditions or under pressure. The shape of the cross section is usually determined by the external and internal forces to be resisted. Pressure tunnels through soft earth must be circular. A horseshoe shape serves well both hydraulically and structurally for firm earth and soft rock.

A good example of a recently constructed power tunnel serves the Apalachia project of the Tennessee Valley Authority.* This structure consists of 35,600 ft of 18-ft diameter lined tunnel, 6000 ft of 20- and 22-ft nominal diameter unlined tunnel, and 2000 ft of 16- and 18-ft steel penstock or liner. The maximum discharge through the tunnel is about 3200 cfs which gives a velocity of 13 fps through the 18-ft diameter section.

Field tests on the completed tunnel indicated the following values of n: for 18-ft-diameter steel pipe, $n = 0.011$ for all velocities between 4 and 13 fps; for 18-ft diameter concrete-lined tunnel, $n = 0.0123$ at 4 fps, and $n = 0.0137$ at 13 fps. For 20- and 22-ft-nominal-diameter unlined rock, $n = 0.0407$ at 2.5 fps through $n = 0.039$ at 4 fps to $n = 0.038$ at 8 fps.

Modern power tunnels of shorter length than the Apalachia tunnel were built by the Tennessee Valley Authority at the Wautaga and the South Holston projects.

6–12. Water Hammer. Water hammer is a phenomenon which is caused by a sudden reduction of the velocity of flow of water in a closed conduit (Fig. 6–17). It is manifested by a series of pressure pulsations above and below normal operating pressure. The magnitude of the pulsations is a function of the rate of stoppage or valve closure and the elastic properties of the pipe material and of the flowing water. Maximum pressure rise will occur if the complete stoppage or closure occurs in less time than it takes the pressure wave to travel from the point of stoppage to a point of relief and return. The rate of travel of the pressure wave happens to be the velocity of sound in fresh water at given temperatures, modified by the elastic properties of the conduit. For water power penstocks the velocity of sound may be taken as 4660 fps and the velocity of the pressure wave, a, as

$$a = 4660 \sqrt{\frac{1}{1 + \frac{kd}{Et}}} \qquad (6\text{–}1)$$

* See *Design of TVA Projects*, Vol. I, *Civil and Structural Design*, TVA Technical Report No. 24 (Knoxville, Tenn.: Tennessee Valley Authority, 1952); and *Trans. A.S.C.E.*, CXIII (1948), 1027–1076.

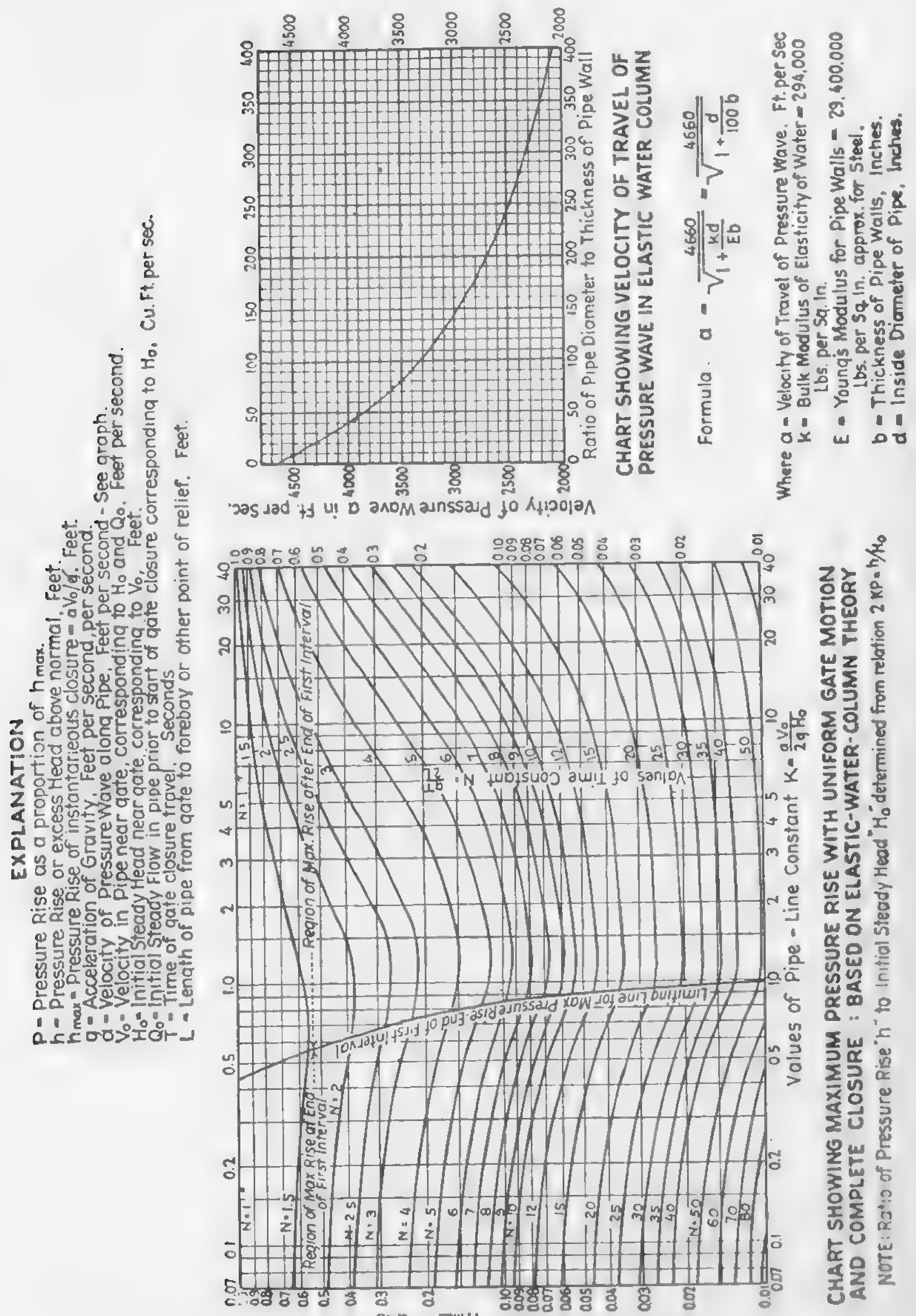


Fig. 6-17 Water hammer chart. (Bureau of Reclamation)

where a = velocity of the pressure wave in feet per second
k = bulk modulus of elasticity of water (294,000 psi)
E = Young's modulus of elasticity of the pipe material
d = diameter of pipe shell in inches
t = thickness of pipe shell in inches

For steel pipe the formula becomes

$$a = 4660 \sqrt{\frac{1}{1 + \frac{d}{100t}}} \tag{6-2}$$

The effect of the elasticity becomes infinitesimal for a penstock concreted in solid rock and a becomes 4660.

The following treatment of water hammer theory is adapted from the work of Ray S. Quick.* The maximum pressure rise, h_{max} (ft), above normal for instantaneous closure is, according to Joukovsky,†

$$h_{max} = \frac{aV_0}{g} \tag{6-3}$$

where V_0 is the initial velocity of flow (fps) in the penstock corresponding to H_0 (ft), the initial steady head; g is the gravity constant. The pressure rise, P, expressed as a proportion of h_{max} is

$$P = \frac{h}{h_{max}} = \frac{h}{\frac{aV_0}{g}} = \frac{gh}{aV_0} \tag{6-4}$$

where h = pressure rise or excess above normal in feet. Other nomenclature is suggested as follows:

K = pipe line constant $= \frac{h_{max}}{2H_0} = \frac{aV_0}{2gH_0}$

L = length of any uniform section of pipe from gate to forebay or any other point of relief

N = time constant or number of $\frac{2L}{a}$ intervals in time of closure $= \frac{aT}{2L}$

* Ray S. Quick, "Comparison and Limitations of Various Water Hammer Theories," *Mechanical Engineering*, XLIX, 5a (May, 1927), p. 524; and Ray S. Quick, *Symposium on Water Hammer*, published under joint auspices of A.S.C.E. and A.S.M.E., 1933.

† N. Joukovsky, "Water Hammer," translated by O. Simin, Proceedings American Waterworks Association, Vol. 24, 1904, p. 341. Original published in Russian and German, *Memoires de l'Académie Impériale des Sciences de St. Petersburg VIII Serie, Classe Physico-Mathematique*, Vol. 9, No. 5, 1898.

Subscript n = designation of the interval measured in steps of $2L/a$ seconds from the start of gate closure when applied to h, P, V, and ϕ.

T = time of gate closure travel in seconds

ϕ_0 = initial gate opening factor where $V_0 = \phi_0\sqrt{H_0}$

ϕ_n = proportion of initial gate factor, ϕ_0, at times designated by subscript n

Σ = a summation of consecutive like terms in steps of $2L/a$ each, such as

$$\Sigma = \frac{h_0}{h_{\max}} + \frac{h_1}{h_{\max}} + \frac{h_2}{h_{\max}} + \cdots + \frac{h_{n-1}}{h_{\max}} = P_0 + P_1 + P_2 + \cdots + P_{n-1}$$
$$= \Sigma P_{n-1}$$

Quick * states: "Gradual gate closure may be considered the equivalent of a series of instantaneous closure movements, each producing a pressure wave proportional to the velocity destroyed and traveling between the gate and forebay; the summation representing the net pressure change at any time. Thus the pressure will continue to rise at the gate in direct proportion to the velocity destroyed until reflection occurs after $2L/a$ seconds to modify the total in the manner expressed by

$$h_n - h_{n-1} = \frac{a}{g}(V_{n-1} - V_n) - 2h_{n-1} \tag{6–5}$$

which designates the rise in pressure during any interval of time of $2L/a$ seconds.

"Then

$$h_n - h_0 = h_{\max} - \frac{a}{g}V_n - 2\Sigma(h_{n-1}) \tag{6–6}$$

"As the pressure rise starts with gate travel $h_0 = 0$ and when full closure is reached $V_n = 0$ so the final pressure rise at the instant of complete closure is

$$h_n = h_{\max} - 2\Sigma(h_{n-1}) \tag{6–7}$$

"Based on the orifice theory

$$V_n = V_0\phi_n\sqrt{\frac{H_0 - h_n}{H_0}} \tag{6–8}$$

"Then combining Eqs. 6–6 and 6–8, eliminating V_n, dividing through by $h_{\max}$ and writing $\dfrac{h_n}{h_{\max}}$ as P_n, there results

$$P_n = 1 - 2\Sigma - \phi_n 2K - \phi_n\sqrt{1 + 2K(1 - 2\Sigma) + \phi_n^2K^2} \tag{6–9}$$

* *Op. cit.*, pp. 521-525.

"When the pipe consists of different diameters and/or thicknesses, the average value of a may be used. The average value may be computed from the formula

$$\text{Average } a = \frac{L_1a_1 + L_2a_2 + L_3a_3 + \cdots L_na_n}{L_1 + L_2 + L_3 + \cdots L_n}\text{"} \tag{6–10}$$

Figure 6–17, as developed by Quick, shows a graphical solution of Eq. (6–9). This figure will serve to solve the majority of practical

FIG. 6–18. Restricted orifice surge tank at the Guernsey Dam power plant. (Bureau of Reclamation)

cases of complete closure and uniform gate motion. The step-by-step method may be developed from the above theory, and often gives more accurate results. However, the theory given here will suffice for preliminary solutions. An example of a solution by Fig. 6–17 follows:

Illustrative Example: Given a pipeline of uniform diameter of 13 ft 9 in., uniform thickness of 2⅜ in., and 573 ft long, carries a flow of 1960 cfs at a velocity of 13.2 fps. The static head is 180 ft and the net head is 167.5 ft.

$$a = \frac{4660}{\sqrt{1 + \frac{165}{100 \times 2.375}}} = 3580 \text{ fps}$$

$$K = \frac{aV_0}{2gH_0} = \frac{3580 \times 13.2}{2 \times 32.2 \times 167.5} = 4.37$$

$$\text{Critical time} = \frac{2 \times 573}{3580} = 0.322 \text{ sec}$$

Assume 16 intervals, then $T = 16 \times 0.322 = 5.125$

$$\text{Check } N = \frac{3580 \times 5.125}{2 \times 573} = 16$$

From Fig. 6–17,

$$P = \frac{h}{h_{\max}} = \frac{gh}{aV_0} = 0.037$$

$$h_{\max} = \frac{aV_0}{g} = \frac{3580 \times 13.2}{32.2} = 1468 \text{ ft}$$

$$h = 0.037 \times 1468 = 54.3 \text{ ft} = 32.4\% \text{ of } 167.5$$

Water hammer problems become much more complicated than is indicated here when branch pipes are involved.*

6–13. Surge Tanks. Surge tanks are essentially forebays near the power unit and are usually necessary parts of medium- and high-head plants. This is particularly true when there is a considerable travel distance between the water source and the power unit. A suddenly reduced load on the turbine causes the governor to close the wicket gates in order to hold the speed steady. This means that water already on its way to the turbine must be stored until an equilibrium of flow conditions can be established. On the other hand, when additional load is required, the governor causes the wicket gates to open and additional water is required to maintain steady speed. The surge tank is therefore a reservoir which furnishes space, immediately available for the acceptance or delivery of water to meet the requirements of load changes. The surge tank also serves to relieve water-hammer pressures within the penstock under conditions of load rejection and acceptance. Three functions are therefore served by the surge tank: (1) flow regulation, (2) water-hammer relief or pressure regulation, and (3) improvement in speed regulation.

Surge tanks are of three common types: (1) simple, (2) restricted orifice, and (3) differential.

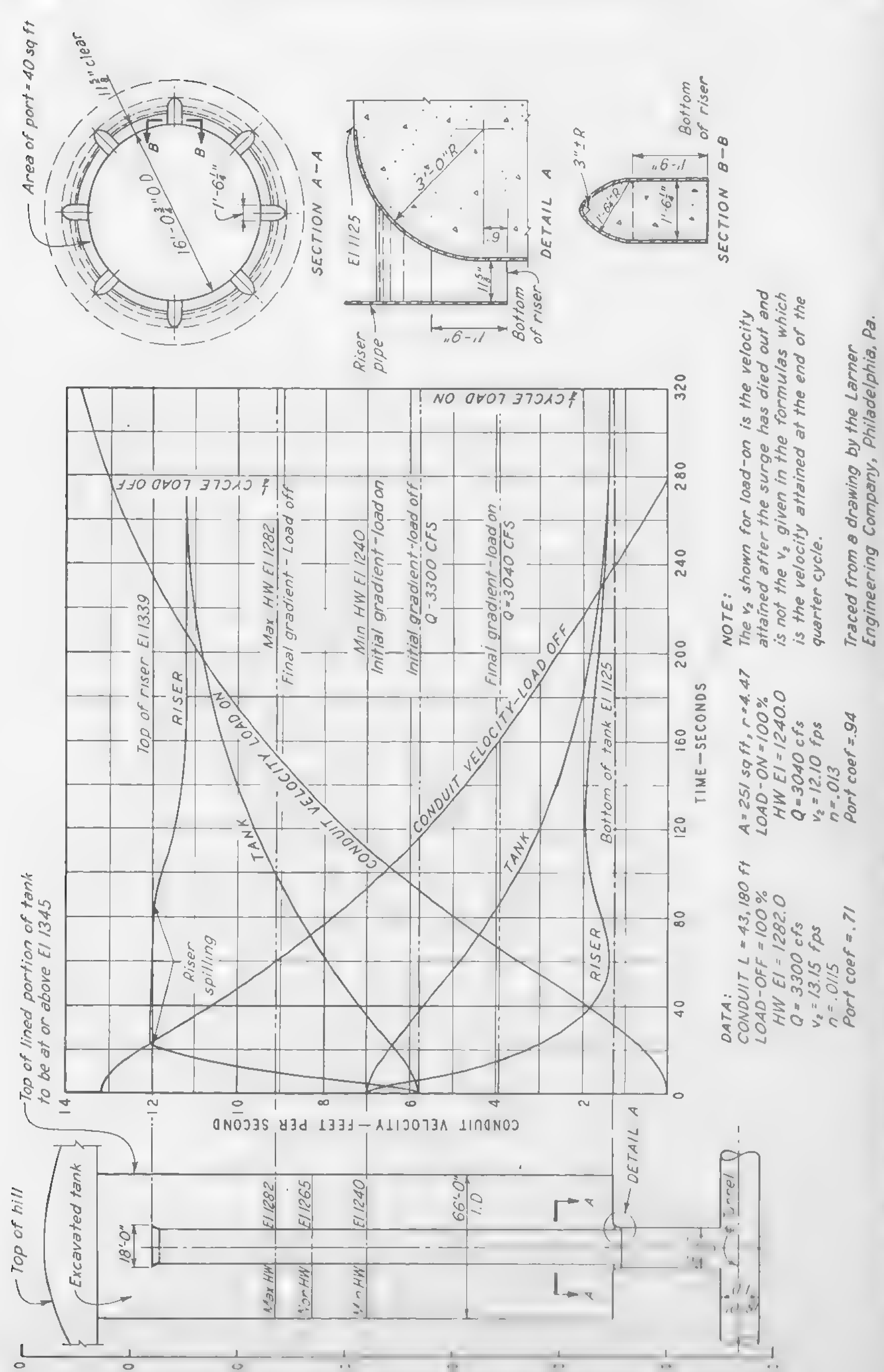
* For the solution of more difficult problems, the student is referred to *Symposium on Water Hammer*, published under the joint auspices of the A.S.C.E. and A.S.M.E., 1933.

The *simple surge tank* acts as a reservoir directly connected to the penstock line in such a manner that water flows into and out of the tank without any appreciable loss of head. It is sluggish in its action and requires a diameter greater than the other two types. Simple surge tanks are rarely used in modern practice, except at installations in which the load changes are small or very gradual.

The *restricted-orifice type* (Fig. 6–18), effects a separation of the water storage function from the acceleration and deceleration functions. An orifice constituting a port is placed between the conduit and the tank. This orifice develops appreciable loss of head by friction when water flows through it into or out of the tank. With a rejection of turbine load, a retarding head is developed by the water flowing into the tank through the orifice. Under conditions of load acceptance by the turbine, the orifice tends to develop an accelerating head in the conduit more quickly than it would be developed in a simple tank. The rapid creation of accelerating and decelerating heads by the restricted orifice tank develops sudden fluctuations of head on the turbine, and thus complicates problems in connection with the governor mechanism. Speed control is also accomplished through the inertia of the rotating parts of the turbine and generator (WR^2). When the governor sensitivity requires the addition of this inertia to the machines, the cost of such addition may preclude the use of a restricted-orifice surge tank.

The differential surge tank (Fig. 6–19) consists of a combination of the simple surge tank and the restricted-orifice surge tank except that an internal riser pipe extends upward from the restricted orifice to near the top of the outside shell. The internal riser is smaller in diameter than the connection to the penstock. This diameter differential results in an annular port which permits water to flow into the tank. The operation of the tank depends upon the area of this port. When additional water is required by the turbine, the water in the internal riser falls rapidly, thus creating an accelerating head rather quickly. The water in the tank flows more slowly through the port and supplies additional water demand. When load is reduced, the water rises rapidly in the internal pipe, thus establishing a retarding head and a differential head on the port, causing water to flow through the port and into the tank. The friction loss dissipates a part of the increased pressure head.*

* For detailed information on surge tanks and their design, the student is referred to the following: *Design of TVA Projects*, Vol. I, *Civil and Structural Design*, TVA Technical Report No. 21 (Knoxville, Tenn.: Tennessee Valley Authority, 1952), Appendix A; George R. Rich, *Hydraulic Transients* (New York: McGraw-Hill Book Co., Inc., 1951); Creager and Justin, *Hydroelectric Handbook* (2d ed.; New York: John Wiley & Sons, Inc., 1950).



(Tennessee Valley Authority)

6-14. Governors. The purpose of a governor is to control speed. A variation in speed will cause a variation in the frequency of an alternating electric current. Constant speed results in accuracy of timekeeping, production machinery operation, and uniform quality of manufactured products. A turbine, and hence its interconnected generator, tends to decrease its speed as the load is increased. Therefore, the operation of a hydro turbine requires the maintenance of a speed as nearly constant as possible. The governor accomplishes speed control

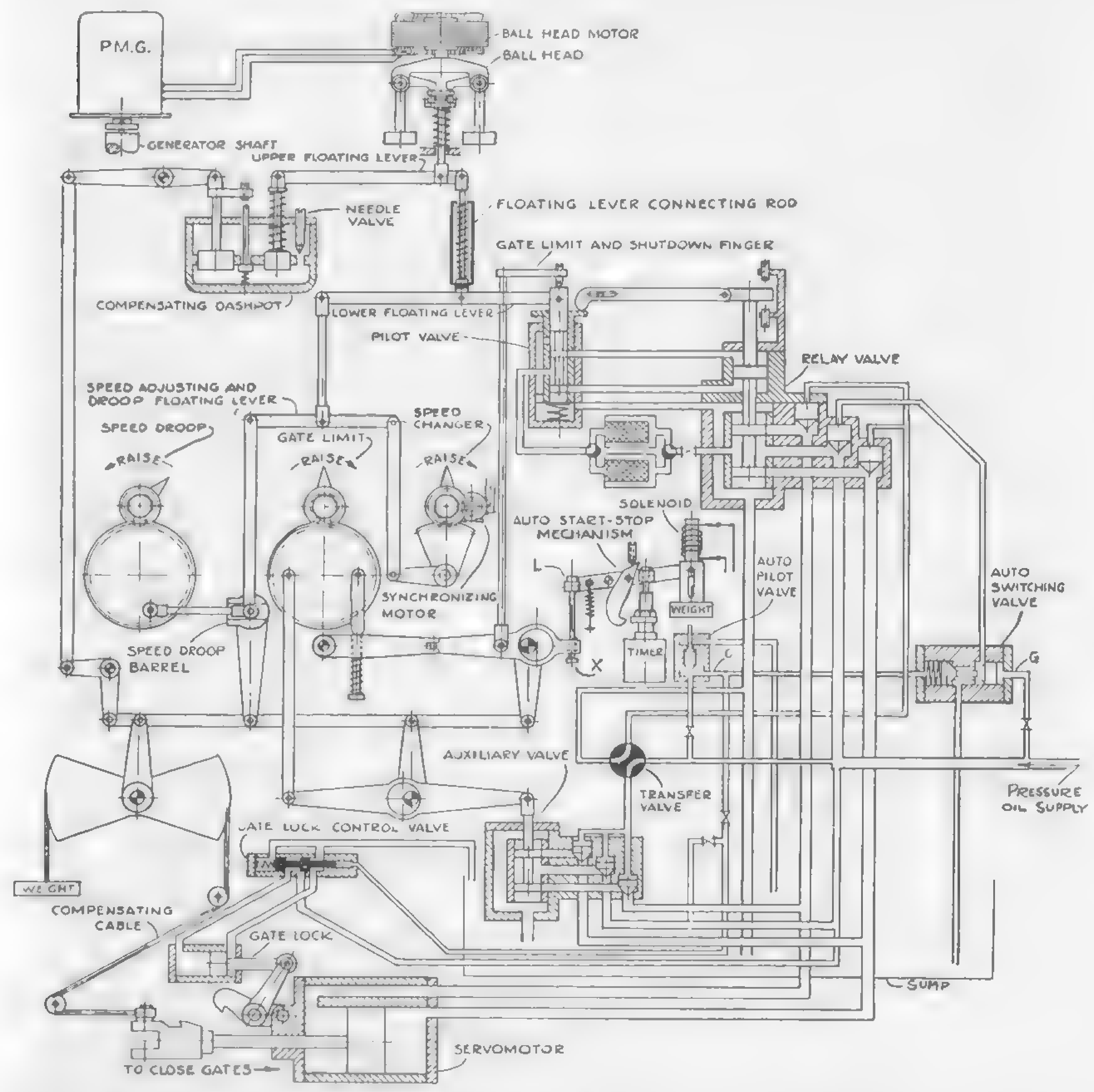


FIG. 6-20. Diagram showing operation of a governor. (Woodward Governor Co.)

by regulating the amount of water allowed to flow through the turbine. It does this by causing a gate or gates of a reaction turbine to open or close or by diverting the water stream away from the buckets of an impulse turbine.

The mechanical operation of a governor is accomplished through three principal elements: (1) a speed-responsive device consisting of

flyballs, usually called the governor head; (2) a power element which serves to change the flow of the energy medium; and (3) a stabilizing or compensating element called the dashpot, which serves to prevent racing of the prime mover by causing the action initiated by the flyballs to be partially counteracted in proportion to the speed of travel of the power-element of the piston. Figure 6–20 shows schematically the principal features of a governor. Another type of governor has been developed which substitutes a standard selsyn motor for the dashpot and a liquid-type tachometer for the flyballs.*

The flyball element is either motor driven or mechanically driven directly by the generator shaft or by belt or gears connecting it to the shaft. Motor-driven flyballs are generally used in modern installations. The current for the driving motor is furnished from either one of two sources: (1) by a permanent-magnet generator, driven directly by the generator shaft, or (2) from the main generator leads. In both instances, the source of power drives a synchronous motor, which in turn causes the flyballs to rotate.

The action of the flyballs is transmitted through a system of floating levers to the dashpot mechanism (element 3) and to a pilot valve. The pilot valve causes oil pressure to be transmitted to the relay or regulating valve. The latter, in turn, causes oil pressure to be admitted on one side or the other of the piston in the servomotor (element 2). The action of the piston opens or closes the turbine gate or gates and thus regulates the flow of the energy medium.

A properly designed governor will be *isochronous,* that is, it will hold the prime mover to a practically constant speed during all changes of loads. Momentary changes of speed from normal will occur each time the load changes, but the normal speed will return immediately. *Speed droop* is the percentage that the normal speed is less at full load than the normal speed at no load. Speed droop adjustment is a feature which must be added to a governor to permit the operation of two or more prime movers operating in parallel and rigidly interconnected.

Governors are of two types: (1) *actuator* and (2) *gate shaft.* The term actuator designates a governor, the valves and control mechanisms of which are separated from the servomotors or power cylinders. Figure 6–21 shows a cabinet for the actuator type installed at Hoover Dam. In the gate-shaft type, the control mechanism is placed directly upon the servomotor. The gate-shaft type is generally used for governor capacities of 60,000 ft-lb and smaller. The actuator type is usually used for capacities ranging from 60,000 to 900,000 ft-lb or more.

* Henry E. Warren, "Precise Turbine Governor," *A.I.E.E. Trans.*, LXVII (1948), pp. 571–74.

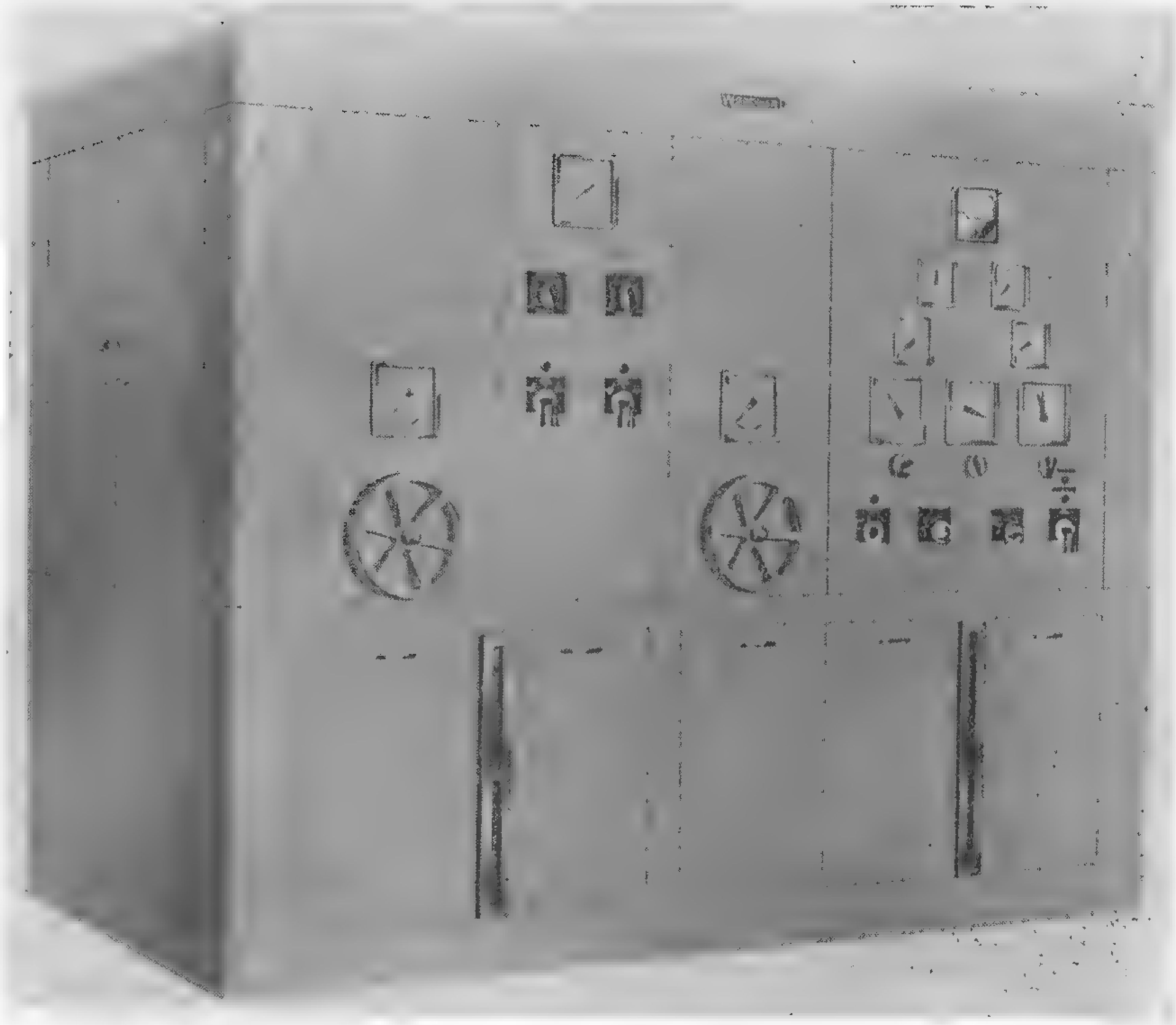


FIG. 6–21. Cabinet for actuator type governor. (Woodward Governor Co.)

Modern governors operate through pneumatic oil pressure systems under pressures ranging from 150 to 300 psi. The pressure used depends upon the amount of energy required to move the turbine gates. The approximate foot-pound rating of a governor for Francis and fixed-propeller-type turbines may be determined by the following formulas:

$$G_1 = \frac{N_s + 10}{10}$$

and

$$G_s = G_1 D_0^2 H_m$$

where G_1 = unit or specific servomotor constant
N_s = specific speed
G_s = servomotor capacity in ft-lb
D_0 = outlet diameter of the runner in feet
H_m = maximum head including water hammer effect

The total servomotor capacity for adjustable-blade propeller runners is usually from two to two and one-half times the values given by the above formulas.

The governor time is the number of seconds required by the governor to move the turbine gates from the closed to the open position or the reverse. The time is limited by the rise of pressure in the penstock and increase of vacuum in the draft tube. For preliminary purposes the interrelation between pressure rise and governor time is assumed to be:

$$h_r = \frac{L \times \text{hp} \times 54}{D^2 \times H^2 \times T}$$

where h_r = per cent pressure rise
L = length of the penstock in feet
hp = rated horsepower
D = diameter of penstock in feet
T = governor time in seconds
H = head in feet

It will thus be seen that the pressure rise varies inversely as the governor time, or that the time varies inversely as the allowable pressure rise. The permissible maximum pressure rise should not exceed 20 per cent.

The design and operation of governors is a highly specialized field. The civil engineer must depend upon the advice and assistance of reputable manufacturers to solve the problems connected with governor selection.

6–15. Cranes. Cranes are devices for raising, transporting, and lowering heavy weights such as machinery, gates, and other heavy equipment. The types used in powerhouses include the overhead traveling crane, gantry crane, and jib crane. Figures 6–22 and 6–23 show the positions of an overhead and a gantry crane as installed in a powerhouse. Table 6–1 shows data on TVA cranes.

The overhead traveling crane (Fig. 6–22) consists of a girder which spans the width of the powerhouse. It travels on rails running longitudinally upon the top of the crane-runway girders. The runway girders are supported by the structural columns. One or two trolleys operate on rails on top of the transverse girder to supply transverse movement. Hoists are mounted on the trolleys to provide for raising and lowering the loads by means of cables and hooks. Lifting beams are sometimes used to distribute the load between the trolleys. Overhead cranes may be operated from a cab or by pull ropes extending to the floor. The minimum clearance between the bridge trucks and the

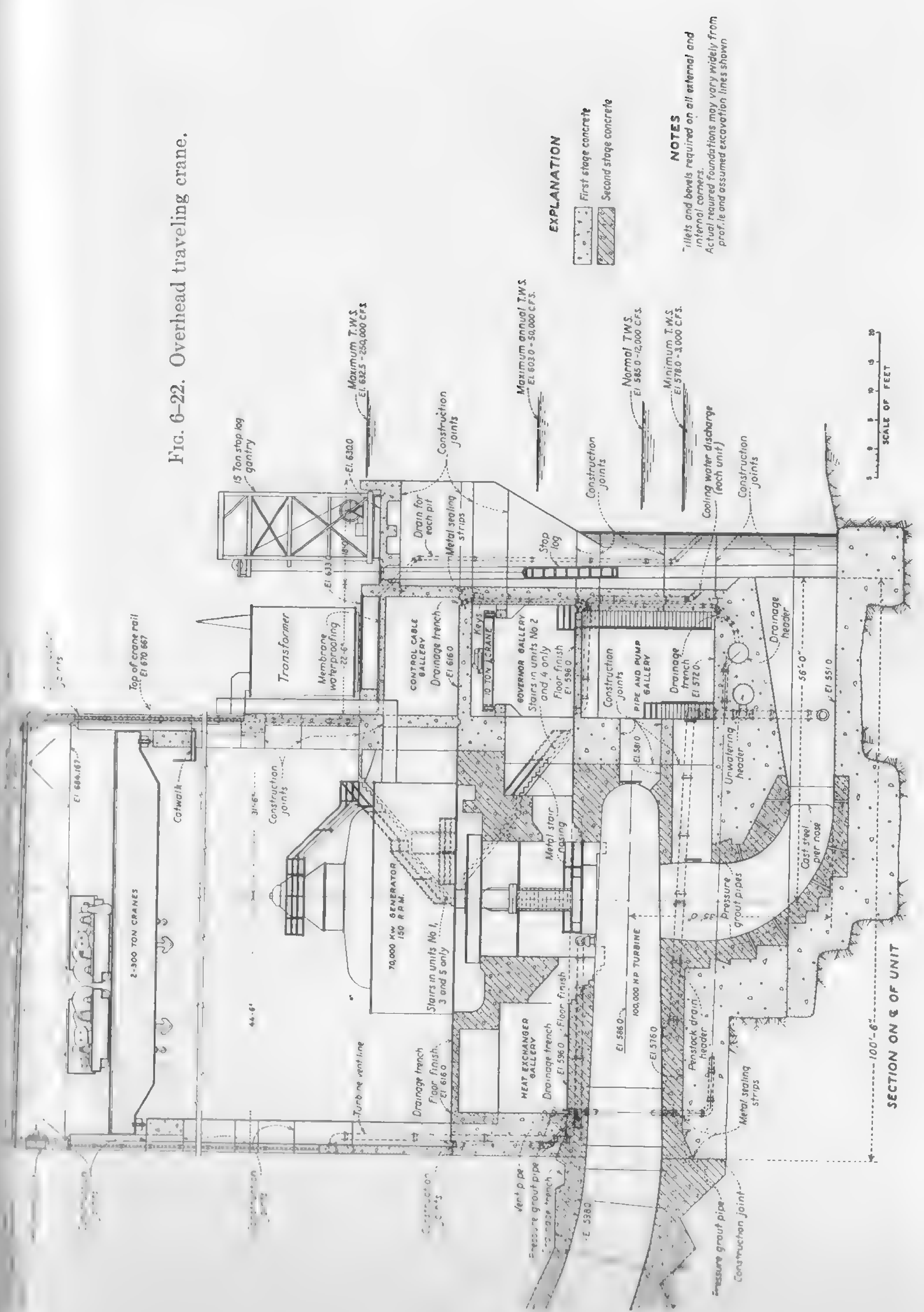


FIG. 6–22. Overhead traveling crane.

wall or column should be 2 in.; the overhead clearance, a minimum of 3 in. Overhead cranes are used in indoor-type powerhouses. The elevation of the crane rail is determined by the clearance required in lifting the longest pieces of equipment. In the Grand Coulee Power Plant of the Bureau of Reclamation (Fig. 6–23), the crane rail is 57 ft 6 in. above the generator floor, and the distance from the top of rail to lowest steel in the roof truss is 15 ft.

A gantry crane (Figs. 6–24 and 6–25) is a handling device mounted on a gantry or a framed structure raised on side supports. Its hoisting operations are similar to the overhead type. The gantry structure is usually mounted on four two-wheeled trucks which travel on tracks laid on the spillway or powerhouse deck. Gantry cranes are used in outdoor or semi-outdoor powerhouses for handling equipment, on the top of

TABLE 6–1

DATA ON TVA CRANES *

Project	Total Capacity, tons	Span, ft-in.	Wheels, No.	Diameter, in.	Trolley No.	Estimated Weight, lb
Overhead						
Norris	250	59–7	16	27	2	445,000
Pickwick	300	58–9	16	27	2	532,000
Guntersville	275	70–1½	16	27	2	600,000
Chickamauga	275	70–1½	16	27	2	680,000
Apalachia	130	40–6	8	27	1	235,000
Ocoee No. 3	100	36–6	8	27	1	225,000
Fontana	300	56–0	16	27	2	550,000
Wautoga	100	41–0	8	27	1	200,000
South Holston	165	40–6	8	30	1	240,000
Gantry						
Wheeler	270	69–0	16	27	2	625,000
Hiwassee	275	63–6	16	27	2	756,000
Watts Bar	225	79–9	16	27	2	871,000
Cherokee	225	65–0	16	27	2	811,000
Kentucky	250	69–0	16	27	2	1,640,000
Fort Loudoun	225	79–9	16	27	2	871,000

* Data from *Design of TVA Projects*, Vol. I, *Civil and Structural Design*, TVA Technical Report No. 24 (Knoxville, Tenn.: Tennessee Valley Authority, 1952).

dams to handle penstock and outlet gates, and also on the downstream deck of a power plant for handling draft-tube bulkhead gates.

FIG. 6–23. Generating room of the Grand Coulee Dam power plant, showing an overhead traveling crane. (Bureau of Reclamation)

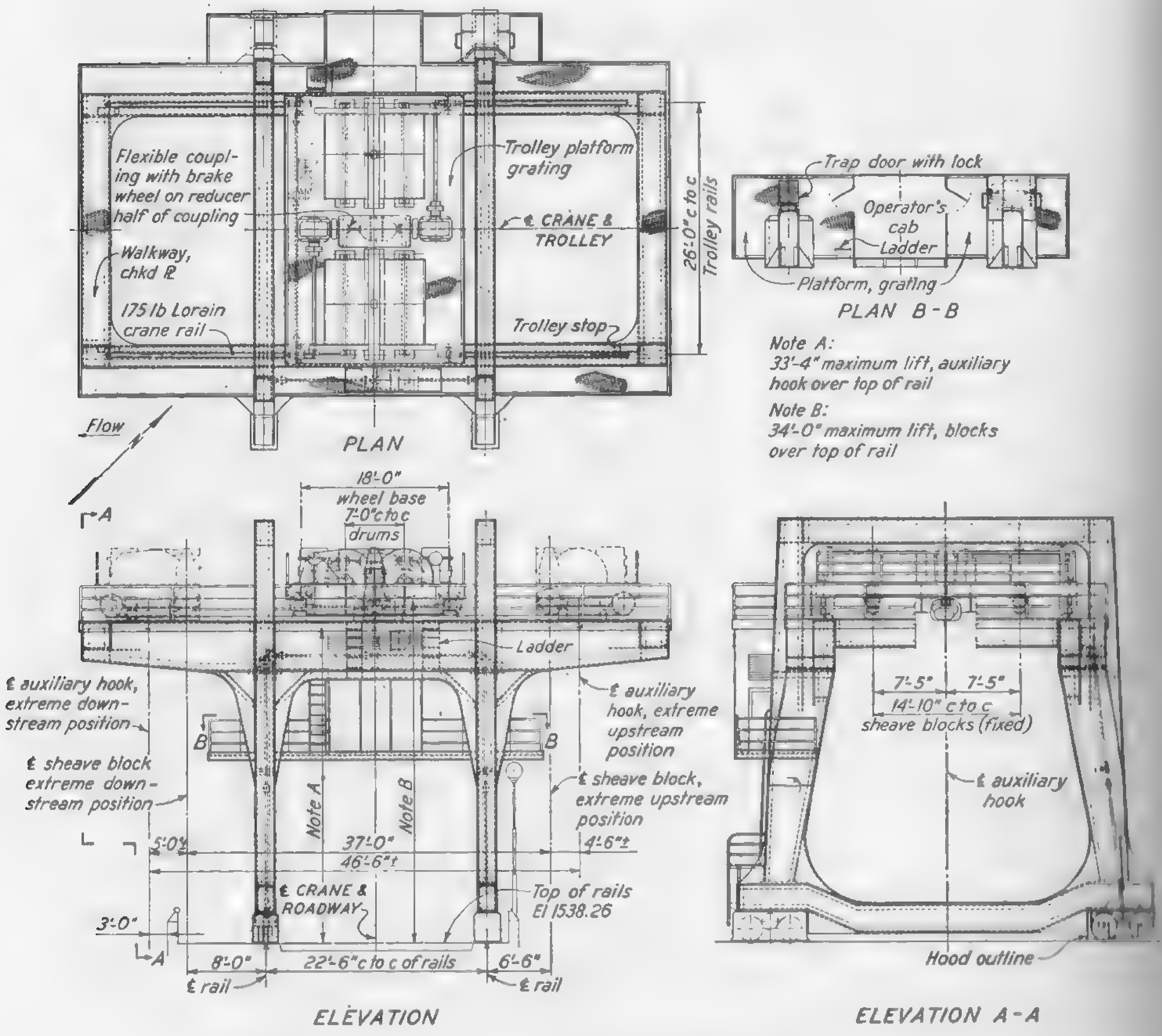


FIG. 6–24. Gantry crane. (Tennessee Valley Authority)

Jib cranes usually consist of a swinging boom provided with a hoisting mechanism. They are frequently attached to the side of a gantry structure and are used to handle intake stoplogs, trashracks, and trash.

Crane specifications. Specification drawings for traveling cranes should give the following information:

- Span, center-to-center of runway rails
- Location of operator's cage
- Building clearances
- Hook approach to each runway rail
- Lift for main and auxiliary hoists
- Size and weight of runway rails
- High and low points of hook travel
- Location and spacing of conductors

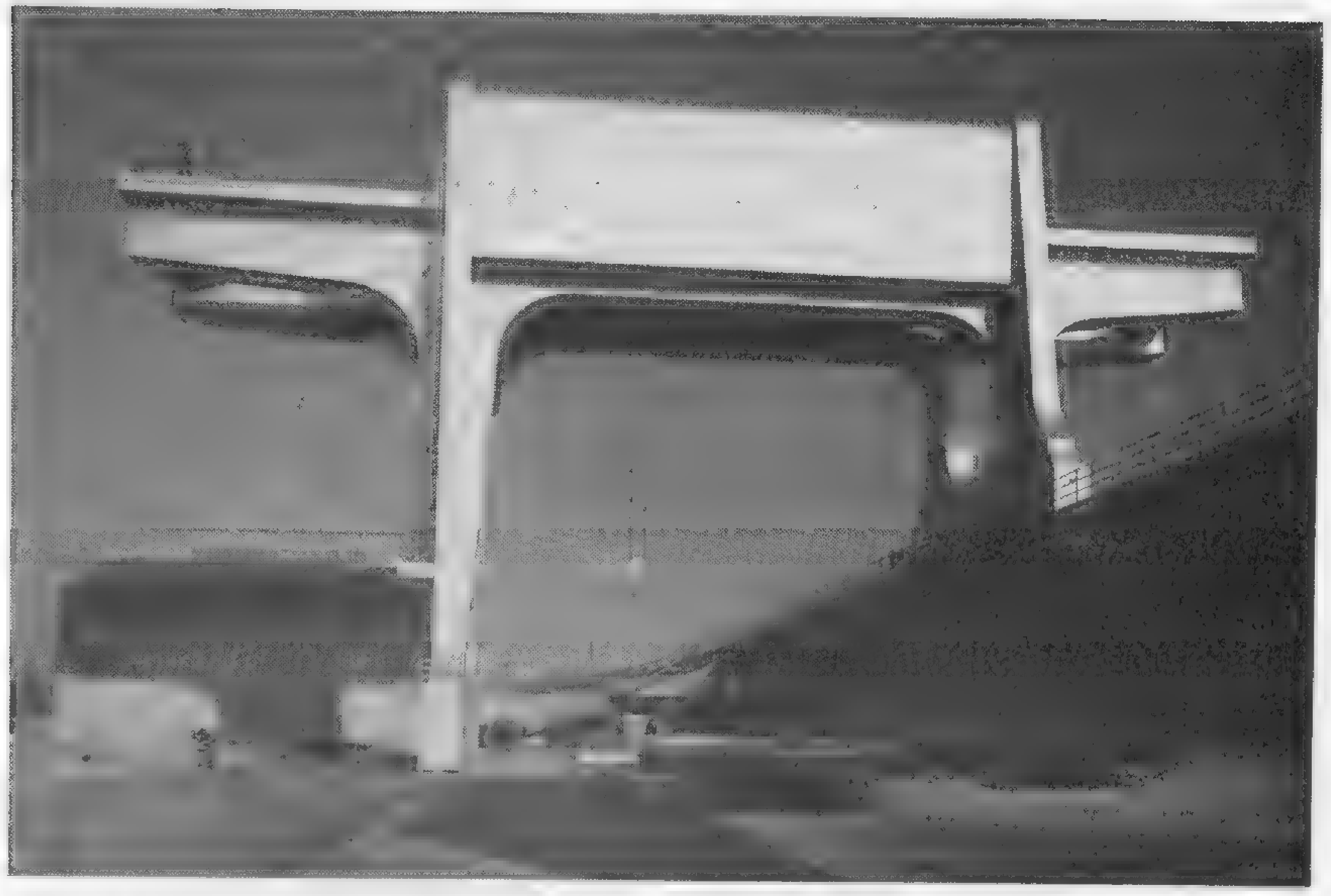


FIG. 6–25. Gantry crane at the Kentucky Dam powerhouse. (Tennessee Valley Authority)

A factor of safety of 5 for all mechanical parts is required.

Base stress for structural parts is 14,000 psi modified for L/r and L/b ratios as required, in which L is the unsupported length, r the radius of gyration, and b the width of the flange.

A mechanical load brake must be provided together with solenoid or thruster-operated brake for the hoist and trolley meters. A bridge travel brake for overhead cranes should be foot-operated and of a mechanical- or hydraulic-actuated type.

6–16. Trashracks. Trashracks (Fig. 6–26) are structures which serve to prevent the passage, through a conduit, of trash and debris of such size as might interfere with gate operation or cause damage to the turbine. They consist essentially of steel bars embedded in and supported by steel shapes. The spacing of the bars depends upon the size of the debris and upon the type of turbine to be protected. Propeller-type turbines have relatively large spaces between the blades and can pass, without damage, larger sized pieces of debris than the smaller-spaced and more intricate-shaped blades of the Francis-type turbine. For example, the main river plants of the Tennessee Valley Authority use propeller-type turbines and the vertical bars of the trashracks are spaced at 70-in. centers. The horizontal bars are spaced at approxi-

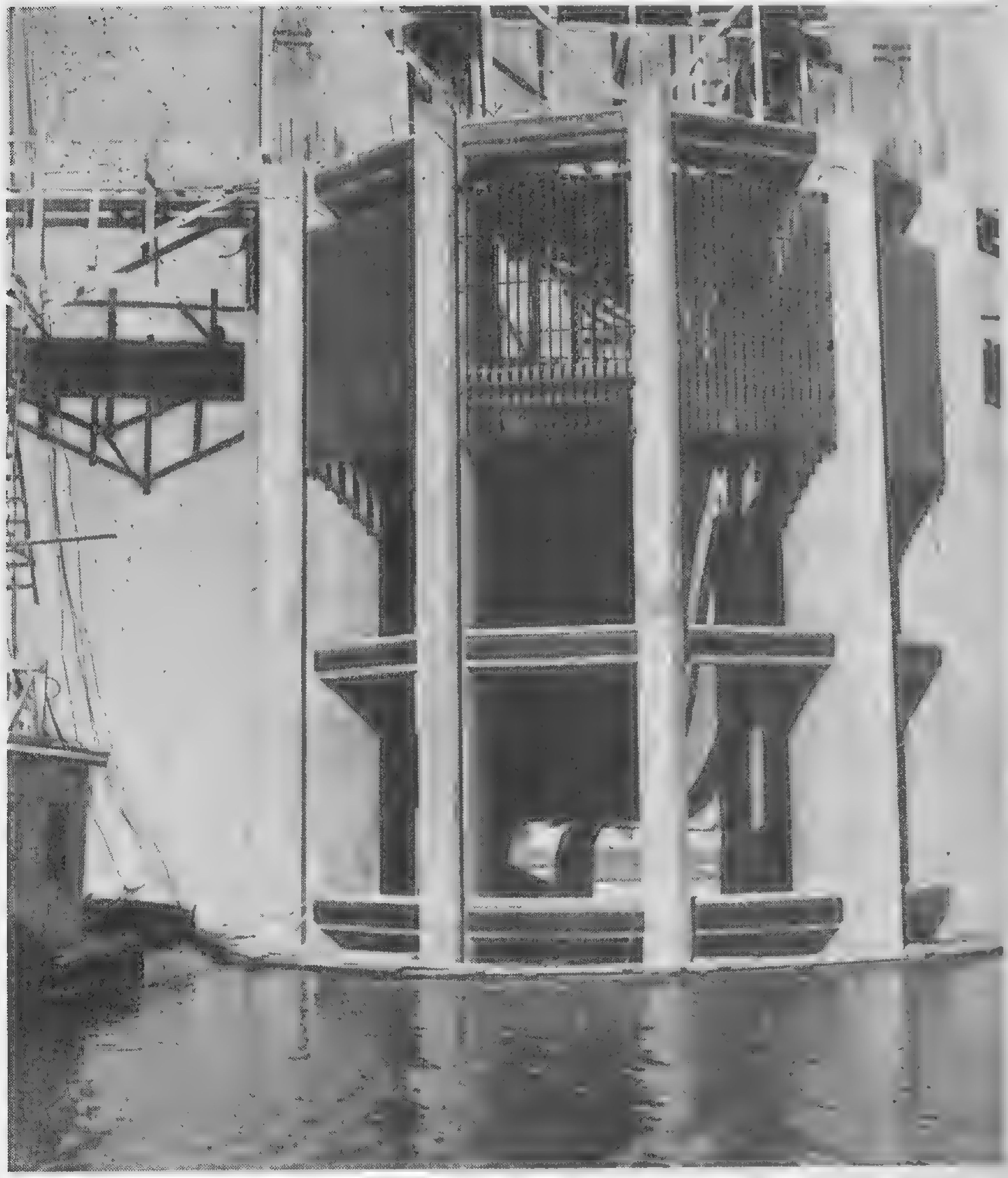


Fig. 6–26. Close-up of trashrack structure, Grand Coulee Dam. (Bureau of Reclamation)

mately 20-in. centers. On the Shasta project of the Bureau of Reclamation (Fig. 6–27), where Francis turbines are installed, the vertical bars are spaced at approximately 6 in. on centers. Eight vertical sections each 12 ft 6 in. long are placed one on top of the other in six tiers, each 8 ft 2⅜ in. wide to protect the total opening having a total width of approximately 49 ft and a total height of 100 ft. The vertical rack sections are supported at top and bottom by reinforced concrete members spaced 12 ft 6 in. on centers, and at the sides by vertical piers spaced 10 ft 8⅝ in. on centers. The vertical piers are arranged in a semicircular manner on a circle having an outside radius of 21 ft 10 in. The rack sections rest in vertical grooves, in the piers and end sections, 7 in. wide and 5½ in. deep on the bearing side. The grooves extend to the top of the dam so that the rack sections can be removed and stoplogs inserted to facilitate cleaning and dewatering operations.

Trashracks should be designed to carry the full water load that may come upon them in case they become completely clogged with ice or debris. The allowable stress in the steel is 18,000 psi. The turbine manufacturer should be consulted in regard to the average velocity of flow through the net area of the racks. For low-head plants this velocity is usually not more than 3.5 fps. For high-head plants, velocities up to 12 fps are used. These values are controlled by economic factors related to the per cent of the total head that may be afforded in loss through the trashrack.

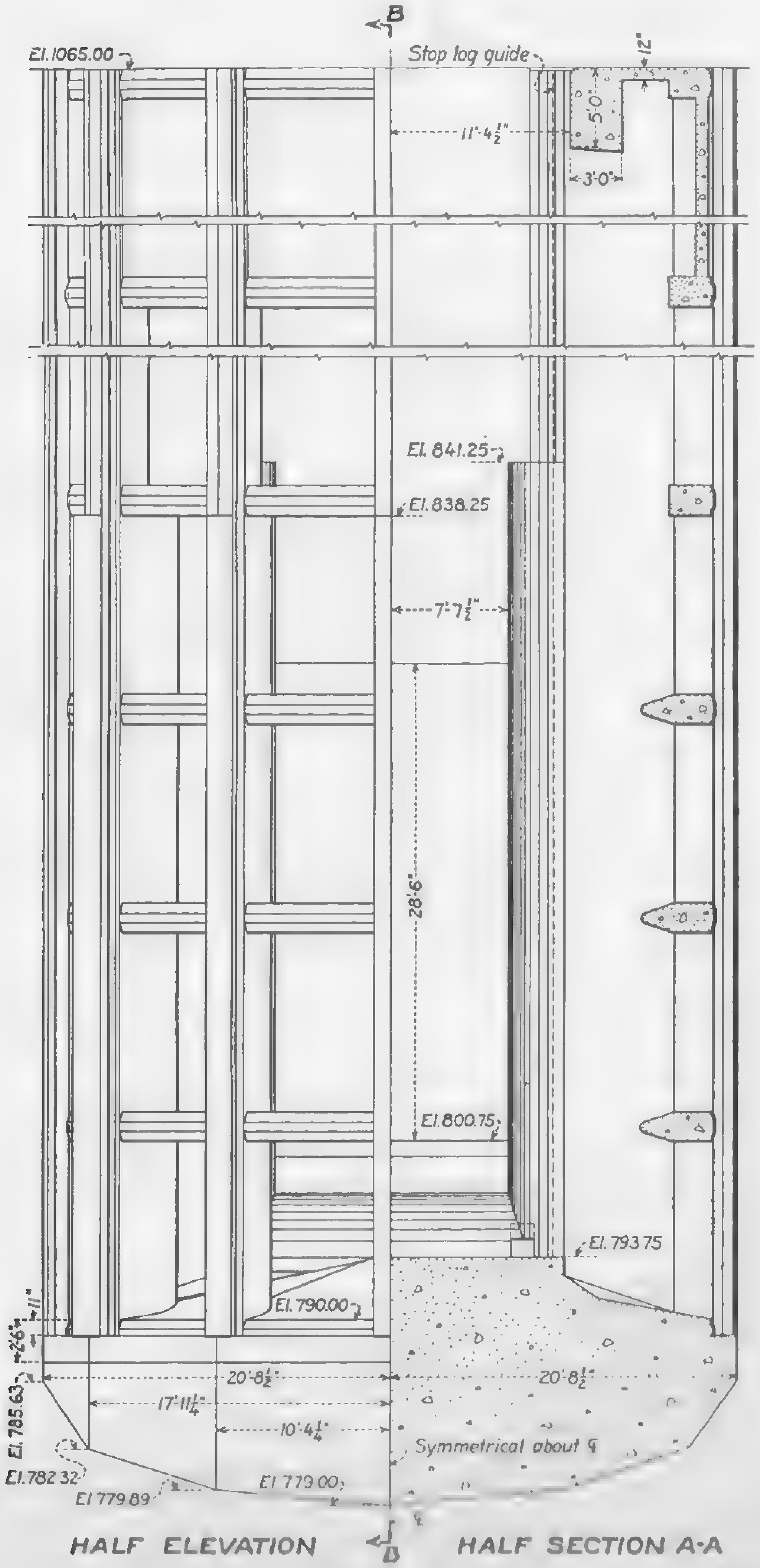
A formula for determining the loss of head through the rack bars was developed by O. Kirschmer as follows:

$$h_r = K\left(\frac{s}{b}\right)^{4/3}\frac{V^2}{2g}\sin A$$

where h_r = loss of head through the rack
s = thickness of the bars
b = clear spacing between bars
$V^2/2g$ = the velocity head
A = the angle of inclination of the rack bars with the horizontal
K is a coefficient depending on the shape of the cross section of the bars.

Values of K are as follows: for rectangular bars $K = 2.42$; for bars rounded at both ends $K = 1.67$; and for circular bars of diameter s, $k = 1.79$.

Trash is removed from racks by hand raking or by mechanical raking. Figure 6–28 shows a sketch of one type of mechanical rake. An illustration of the value of a mechanical rake is shown in Fig. 6–29.



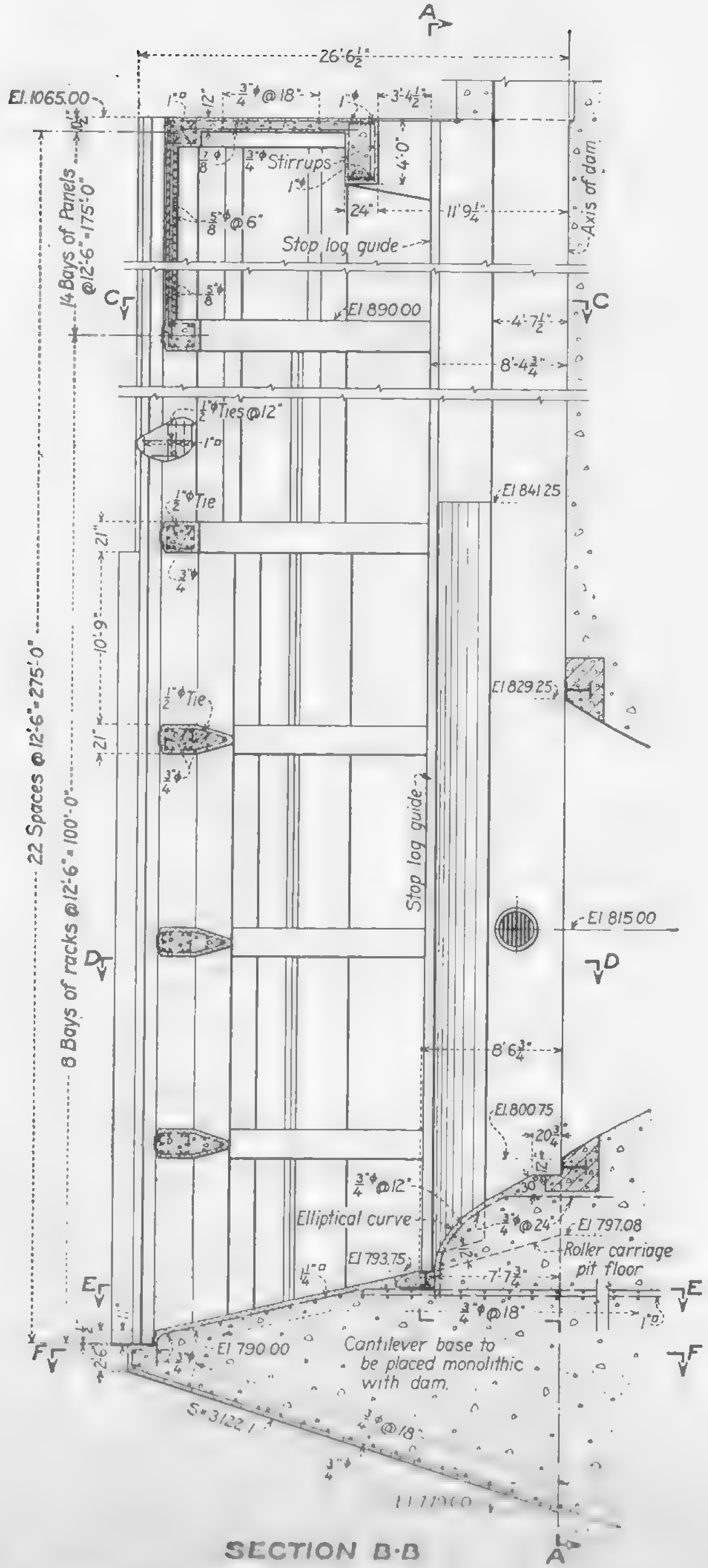


FIG. 6-27A, B. Details of trashrack structure, Shasta Dam. (Bureau of Reclamation)

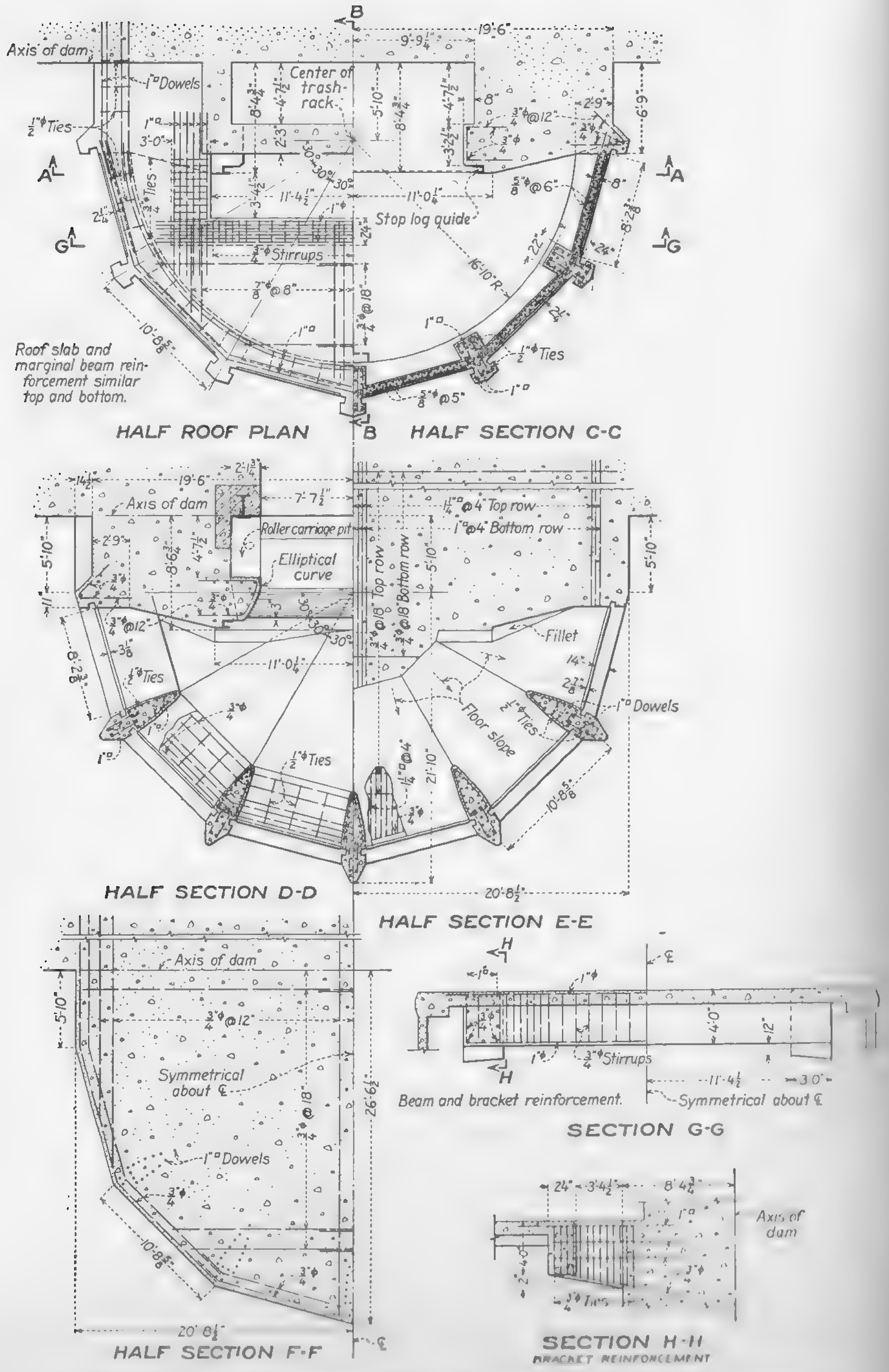


FIG. 6-27C. Details of trashrack structure, Shasta Dam. (Bureau of Reclamation)

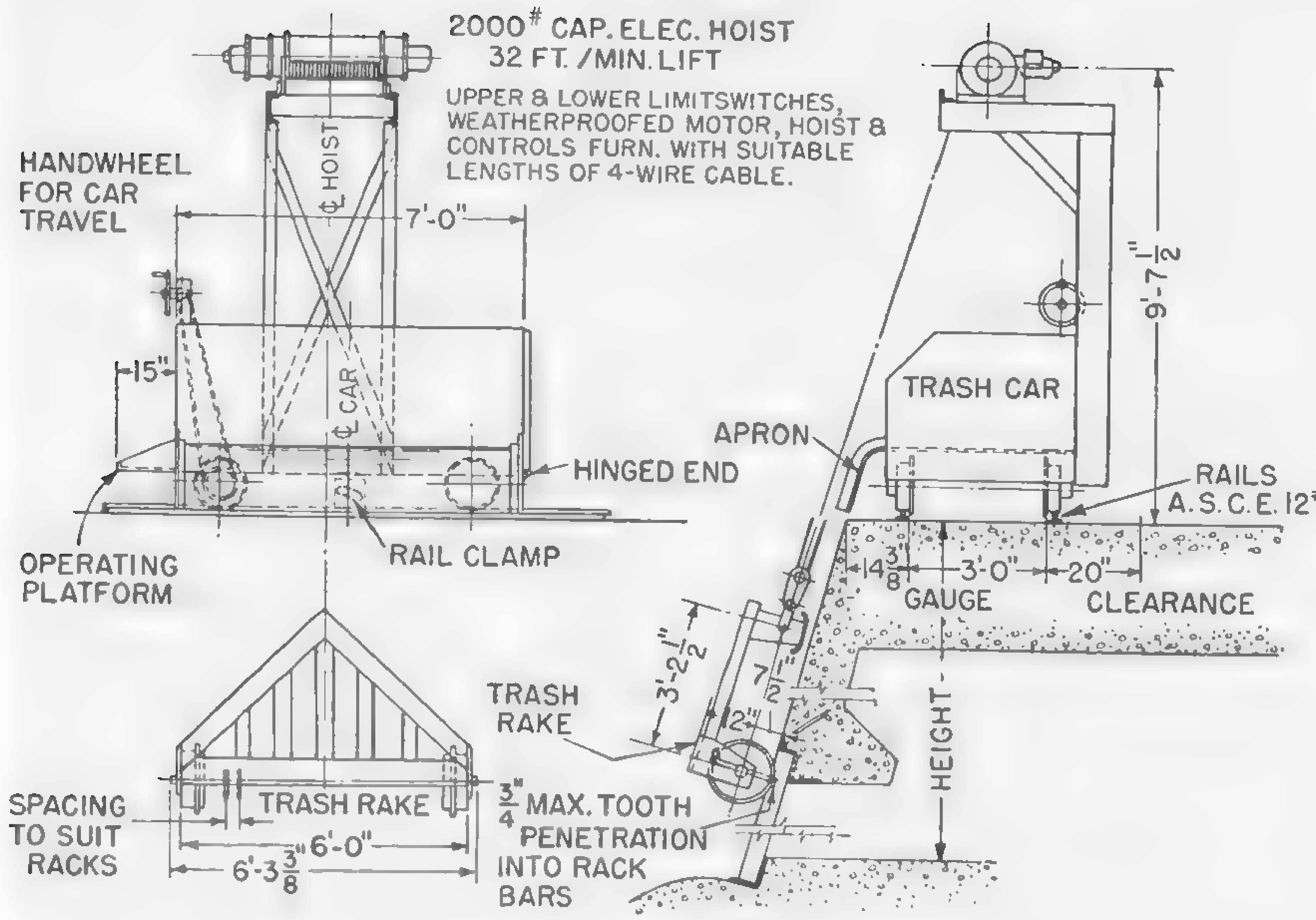


FIG. 6-28. Trashrack and rake. (S. Morgan Smith Co.)

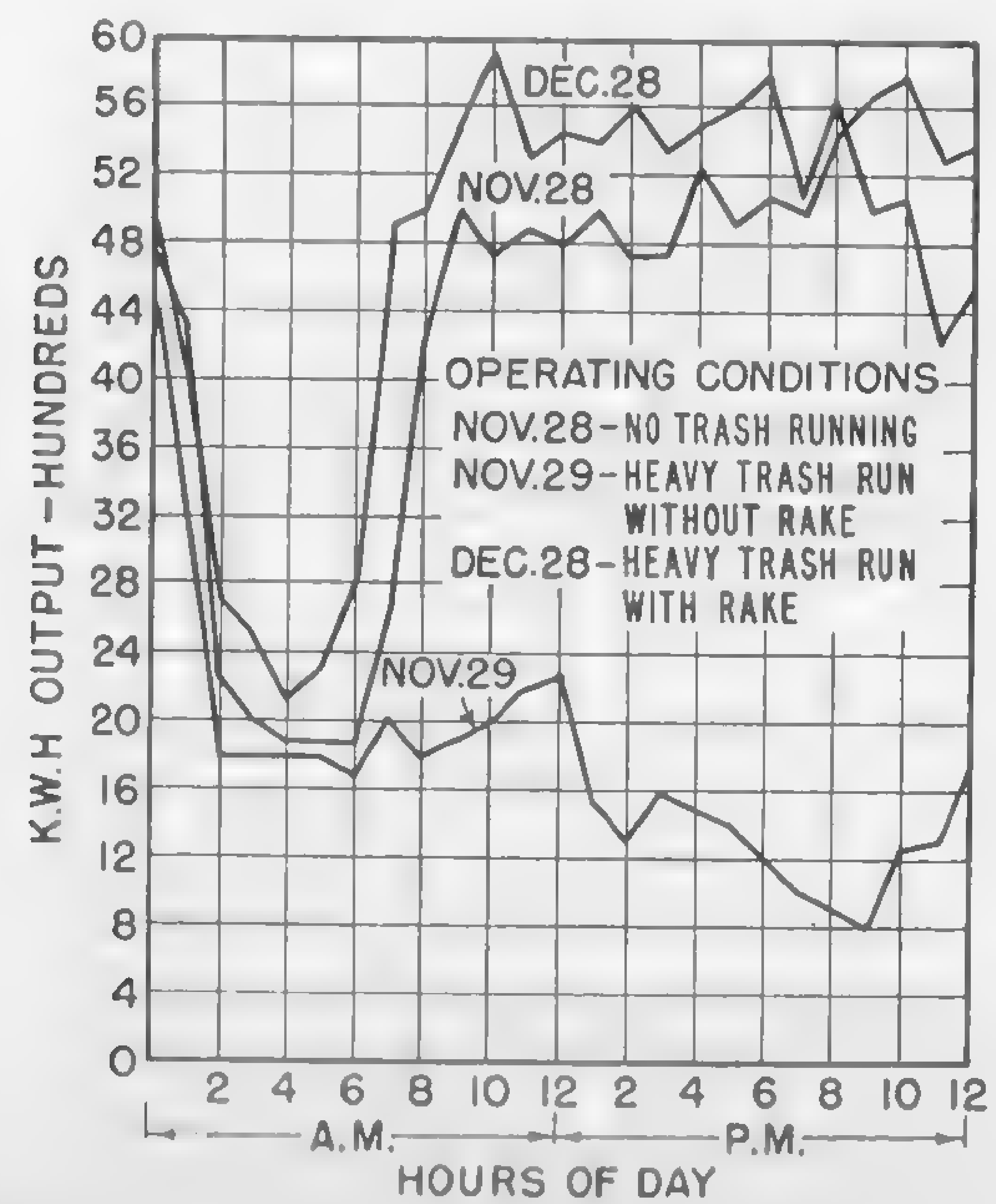


FIG. 6-29. Effect of trash removal on the output of a power plant. (S. Morgan Smith Co.)

In climates where an installation is subject to frazil ice troubles, readily removable trashracks are advantageous. Frazil ice usually forms at a time when little trash is running so that removal of racks for ice clearing will not endanger the turbines. Rakes will not keep racks clear of ice. They are used to break the surface ice when being lowered and to remove it on the upward cleaning stroke.

CHAPTER 7

ECONOMIC ASPECTS OF HYDROELECTRIC DEVELOPMENT

7–1. Introduction. For every hydroelectric development there is a certain combination of size of dam, reservoir, conduits, and installation capacity which will produce the lowest power production cost. The particular combination can be arrived at only after several trials embodying various designs and estimates of cost. The present-day trend in hydro installations does not necessarily require or permit the employment of the minimum production cost installation; rather, the final design is based upon the best utilization of the water resources of the stream. However, every installation should fall within the limits of economic justification.

The installation capacity often can be determined by the multiple-purpose aspects of water resources development. Power may be a secondary component of a given development. The installation capacity is frequently dependent upon the by-product values produced by the power, which can only be generated if the primary purposes of the project are served first. Examples of such primary purposes are irrigation, navigation, and flood control.

Since power is a salable product and is one of the few water-resource commodities which produce monetary returns, it sometimes becomes a "whipping boy" in matters of allocation of cost in multiple-purpose projects. There is sometimes a tendency to slant cost allocation in such a way that power costs are reduced below their actual amount in order to show inflated benefits from one or more of the other components. On the other hand, power revenues may be contemplated for use in subsidizing noneconomic components which may be incorporated in a multiple-purpose development.

Dependable electric generating plants must have reserve capacity available for providing service during periods of breakdown or outage due to failure of or repairs on mechanical equipment or transmission systems. Reserve capacity should also be considered in planning for future growth of load. The amount of reserve capacity required will depend on individual conditions. Factors to be considered include reliability of service, the load factor, and degree of interconnection among others. The range of reserve capacity may be from 10 to 100 per cent. It is not possible to state an average value.

7–2. Allocation of Costs. In multi-purpose developments where several functions including hydro power are to be served, it is essential that the actual cost, and hence the economy, of each function be determined as closely as possible. This practice tends to reveal the noneconomical justification of some of the components and avoids penalizing the justifiable components. Such practice also tends to disclose the extent of subsidization of certain other components. The problem is essentially one of proper apportionment of the total cost of jointly used facilities such as dams, reservoirs, and water conduits. The *proportionate-use-of-capacity method* is recommended as affording a minimum of arbitrariness and a maximum of equity. By this method the total cost of jointly used facilities is allocated to the various components in proportion to the amount of water and storage space required to serve each component or to which each component is entitled. The simplest application of the method consists of allocating to each particular component a certain portion of the total volume of the reservoir, then the cost chargeable to each component is in proportion to the total cost. If dead storage is involved, its volume should be prorated to such components as require its existence. Instances of overlapping utilization of storage, such as the retention of flood water for subsequent use for irrigation and power development, introduce some complications. The question of head utilization may also complicate the problem.*

7–3. Economic Justification. Every hydroelectric project regardless of the source of construction and operating funds should constitute a good business enterprise. In other words, the financial return should be commensurate with the investment, or, at least, every project should be self-liquidating. The test for the economic justification of a new project must necessarily be based upon estimates of benefits versus estimates of cost. Benefits consist of the net annual monetary return which can be expected from the sale value of all of the net power produced. Cost includes the sum of all of the items of expense incurred as a result of the construction of the project. All estimates should be thoroughly inclusive and completely realistic. They should also be adjusted, if necessary, to represent conditions prevailing at the time the construction of the project is ready to begin. The ratio of annual net benefits to annual net costs should be at least 4 to 3 when an early realization of benefits is expected, and not less than 2 to 1 when it is anticipated that benefits will develop slowly. An example of the latter situation would

* For further information on this subject the student is referred to *Tentative Report with Appendix of the American Society of Civil Engineers Committee on Cost Allocation for Multiple Purpose Water Projects* (New York: American Society of Civil Engineers, 1948).

be if the site of the power plant were in a very remote region, so that the development of a market might be considerably delayed.

There are many other considerations entering into the determination of economic justification of water power projects. The Federal Power Commission in its rules and regulations has established certain criteria which affect rates to be charged for power. Also, Acts of Congress specify certain procedures which must be followed by various federal agencies.*

7–4. Total Annual Unit Cost. The total annual unit cost of power production is based upon two components: (1) fixed cost and (2) operation and maintenance cost, sometimes called production expense.

The fixed charges include three elements: (1) interest or cost of money, (2) taxes and insurance, and (3) amortization or depreciation. The fixed charges should include, in addition to interest, additional costs such as for the placing of loans and for the cost of levying and collecting taxes. Interest rates may vary from year to year and may also be dependent upon whether the project is speculative or whether it is being promoted by a utility with a sound credit rating. Public agencies other than federal may finance a project through the sale of bonds which includes an amortization arrangement to pay off the obligation in principal and interest over a period of from 25 to 40 years. At the end of the period, the promoting agency owns the project free of debt. Fixed charges then approach zero. Private companies use various ways of financing, such as the sale of stock or bonds. A fair rate of return to stockholders must also be provided in private utility financing.

Taxes must be provided for in private financing, and payments in lieu of taxes should be provided for in public developments. Consumers of electric energy whether generated by a public or private utility should be treated alike in the establishment of rate structures.

Insurance, underwritten by a separate company, may or may not be carried on a project, but in either event provision should be made for the risks involved in the possible destruction of property through fire, windstorms, floods, and the like. Liability insurance may be carried on buildings, and equipment and workmen's compensation may or must be carried depending on state laws.

Amortization may be defined as the annuity portion of sinking fund depreciation or as the uniform annual payment to effect capital recov-

* For more specific recommended national policies, the student is referred to *Principles of a Sound National Water Policy*, Engineers Joint Council National Water Policy Panel, July, 1951, Appendices 5 and 9; and *Proposed Practices for Economic Analysis of River Basin Projects*, Report of the Subcommittee on Benefits and Costs to the Federal Inter-Agency River Basin Committee, May, 1950.

ery of the difference between first cost and salvage value. Amortized cost is an accounting concept of depreciation. E. L. Grant states: "From the viewpoint of accounting, the cost of an asset is a prepaid operating expense to be apportioned among the years of its life by some more or less systematic procedure. The controversial questions here are what the estimated life should be and how the apportionment should be made. It should be emphasized that it is cost, not value that is apportioned in orthodox accounting." *

Amortization therefore consists of the making of actual or hypothetical payments toward the liquidation of the investment. The straight line method or the sinking fund method may be used. The straight line method involves provision for the repayment of the investment in equal annual amounts over the assumed period of life of the project. The sinking fund method involves the actual or hypothetical payment of an annual sum which, if invested at compound interest, will at the end of the life of the project produce the amount of the original investment plus the invested sum. In computing the annual payment, A, by the sinking fund method, the following formula is used

$$A = \frac{Cr}{(1+r)^n - 1}$$

where C = the first cost
r = the interest rate
n = the estimated life

Table 7–1 shows the estimated average useful life of various power-producing facilities.

Note that the estimated life of the components of a hydro plant is quite variable. The National Water Policy Panel of Engineers Joint Council recommends that the period of amortization should not exceed 50 years, and that either of the two following procedures be followed in connection with those components which have a shorter estimated life than the period of amortization:

1. Set up a higher annual rate of amortization for the shorter lived components.
2. Provide a separate allowance for interim replacements.

"The recommended amortization requirement makes unnecessary any allowance for depreciation beyond the provision for interim replacements." †

* E. L. Grant, *Principles of Engineering Economy* (3d ed.; New York: The Ronald Press Co., 1950), p. 177.

† *Principles of a Sound Water Policy*, Engineers Joint Council National Water Policy Panel, July, 1951, p. 212.

TABLE 7–1

AVERAGE USEFUL LIFE, ELECTRIC UTILITIES *

Item	Life Years
Average life, typical system, steam-generated power	31
Average life, typical system, hydroelectric power	40
Steam Production	
Structures and improvements	50
Boiler plant equipment	28
Engines and engine-driven generators	30
Turbogenerator units	30
Accessory electric equipment	28
Miscellaneous power plant equipment	28
Hydraulic Production	
Structures and improvements	75
Reservoirs, dams, and waterways	150
Waterwheels, turbines, and generators	35
Accessory electric equipment	35
Miscellaneous power plant equipment	35
Roads, railroads, and bridges	100
Transmission Plant	
Structures and improvements	45
Station equipment	28
Towers and fixtures	50
Poles and fixtures	33
Overhead conductors and devices	50
Underground conduit	75
Underground conductors and devices	40
Roads and trails	60

* Data taken from *Bulletin F, Income Tax Depreciation and Obsolescence, Estimated Lifes and Depreciation Rates*, Bureau of Internal Revenue (Washington, D. C.: Government Printing Office, 1942), p. 61.

7–5. Cost of Hydro Installation. A hydroelectric plant usually involves some or all of the following items: reservoir, dam, waterways, land, water rights, structures, and equipment. Each development is a special case largely because of topographic and geologic conditions so that no general distribution of the cost of the various components can be made safely. However, according to studies made by the Federal Power Commission, with the average distribution for heads under 100 ft, costs are distributed approximately as follows: dams and reservoirs, 47 to 55 per cent; equipment, 18 to 28 per cent; structures, 10 to 13 per cent; and land, 13 to 17 per cent. For heads between 100 and 500 ft, cost distributions are approximately: dams and reser-

voirs, 54 to 63 per cent; equipment, 14 to 25 per cent; structures, 10 to 12 per cent; land, 11 to 14 per cent.

The Federal Power Commission published a report in 1948 which contains an analysis of total cost for 984 private utility plants, most of which were built prior to 1930.* The results of this analysis are shown in the lower left hand part of Fig. 7–1. In the March and April, 1951,

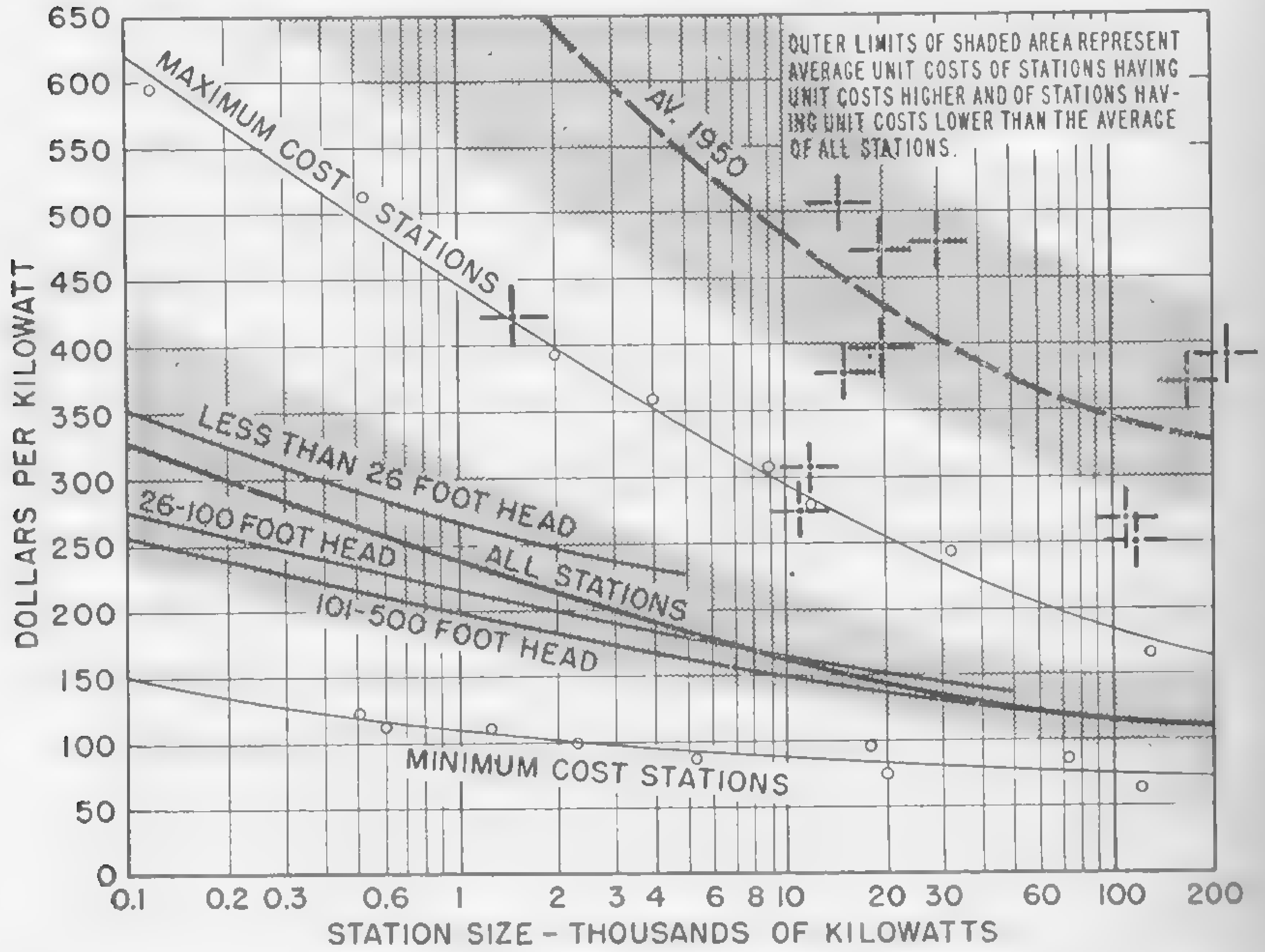


Fig. 7–1. Range of total costs per kilowatt of installed capacity. (C. K. Willey, *Midwest Engineer,* March, 1951, p. 27)

issues of the *Midwest Engineer* C. K. Willey draws a comparison of the estimated 1951 costs with the actual cost of the plants analyzed by the Federal Power Commission. The results of these studies are shown in the upper right hand portion of Fig. 7–1. Mr. Willey states: "The crosses [in the figure] are plots of the cost, estimated or actual, of plants which our company [Harza Engineering Company] has studied within the past three years and which are economical to develop." †

* *Electric Utility Cost Units—Hydroelectric Generating Stations,* U. S. Federal Power Commission (Washington, D. C.: Government Printing Office, 1948).

† C. K. Willey, "Production Economics in Hydro Power," *Midwest Engineer* (March, 1951), p. 12; (April, 1951), p. 10.

Improvements in earth moving machinery, excavation equipment, and other heavy equipment have tended to offset rising prices, while higher labor and material costs have tended to change design. As an example, the high cost of labor and materials has tended to reduce use of reinforced concrete for dams in favor of gravity concrete structures.

To arrive at a unit cost per kilowatt of installed capacity, it is necessary to add together the estimated cost of the separate elements allocable to the power component, such as the land, dam, reservoir, penstocks, and powerhouse equipment, and divide by the proposed installed capacity. Since the capacity of the several elements control the installed capacity, a decision must be reached in regard to the available capacity as indicated by the stream flow. Since Fig. 7–1 is based on actual experience, it may be useful as a guide for economical unit cost of installation and also for preparing problems for student instruction in power economics.

The cost of intakes, conduits, powerhouse equipment, tailrace, substations, and all other works required for the production and distribution of power are more likely to vary roughly in proportion to the installation than are the costs for the dam and reservoir, railway and highway relocation, and other items not directly related to the installed capacity. The term "incremental cost" is sometimes applied to the cost of those features which are directly related to the number of kilowatts installed. Thus, if the total cost of a 50,000-kw plant is $370 per kw and the cost of equipment and power structures is 30 per cent, or $111, it is reasonably safe to assume that the total cost of increasing the installation to 100,000 kw would be $111 × 950,000, or $5,550,000.

The term incremental costs as applied to multiple-purpose projects is defined by the National Water Policy Panel of Engineers Joint Council as "By '*incremental costs*' are meant the costs incurred solely because of incremental components, being components which are either added to or the inevitable result of primary components and could not or would not be included except for the latter." *

7–6. Fixed Cost, Hydro Plants. When the unit cost of installation has been determined, the annual charges per kilowatt of installed capacity may be determined. This requires an investigation of the money market, tax and insurance rates, and decisions in regard to rates of amortization or depreciation. Table 7–2 shows the range and average values of rates used for the determination of fixed costs based on the capital cost of the investment.

* *Principles of a Sound National Water Policy,* Engineers Joint Council National Water Policy Panel, July, 1951, p. 202.

TABLE 7-2

Fixed Costs (Charges)	Range %	Average %
Cost of money	5.0– 8.5	6.5
Taxes and insurance	0.5– 1.5	1.0
Amortization, depreciation	1.0– 4.0	2.0
Total	6.5–14.0	9.5

7-7. Production Costs, Hydro Plants. The cost of production or maintenance and operation, exclusive of fixed charges, includes supplies, expenses, supervision and engineering, maintenance, and station labor. The 1950 report of the Federal Power Commission states:

"For hydro station average production expenses (exclusive of fixed charges) per kilowatt of capacity for 5,000, 25,000 and 100,000 kilowatt stations are \$6.10, \$2.70 and \$1.80 respectively. At 60 per cent annual plant factor such costs are equivalent to 1.16 mills, 0.51 mill and 0.34 mill per kilowatt hour respectively. As station size increases beyond 100,000 kilowatts, there is little further decrease in average unit expenses." *

Figure 7-2 shows these values as well as those for a range in size from 100 to 200,000 kw.

Illustrative Example: Determine the unit cost per kilowatt-hour in a hydro plant which has an installed capacity of 100,000 kw for plant factors of 50 and 60 per cent.

From Fig. 7-1 the average unit cost of installation (1950) is \$345.00; fixed charges, $0.095 \times 345 = \$32.775$. Production: at 100 per cent load factor, $24 \times 365 = 8760$ kwh per installed kilowatt; at 50 per cent load factor, 32775 mills $\div$ 4380 = 7.48 mills per kwh for fixed charges. Total unit cost per kilowatt-hour, $7.48 + 0.41$ (production expense $= 1800/4380 = 0.41$) = 7.89 mills per kwh; at 60 per cent plant factor total unit cost is 6.56 mills per kwh.

7-8. Transmission Liability. The term *transmission liability* is sometimes applied to hydro plants. It implies that hydro plants, which are usually located at distances relatively remote from load centers, are subject to increased annual costs incurred by the necessity of transmission lines and reduced electric energy dissipated by the current flowing through transmission systems. If an alternative steam plant could be located closer to the load center than a proposed hydro plant, the penalties arising out of transmission costs and losses must be assessed

* *Thirtieth Annual Report,* Federal Power Commission (Washington, D. C.: Government Printing Office, 1950), p. 34.

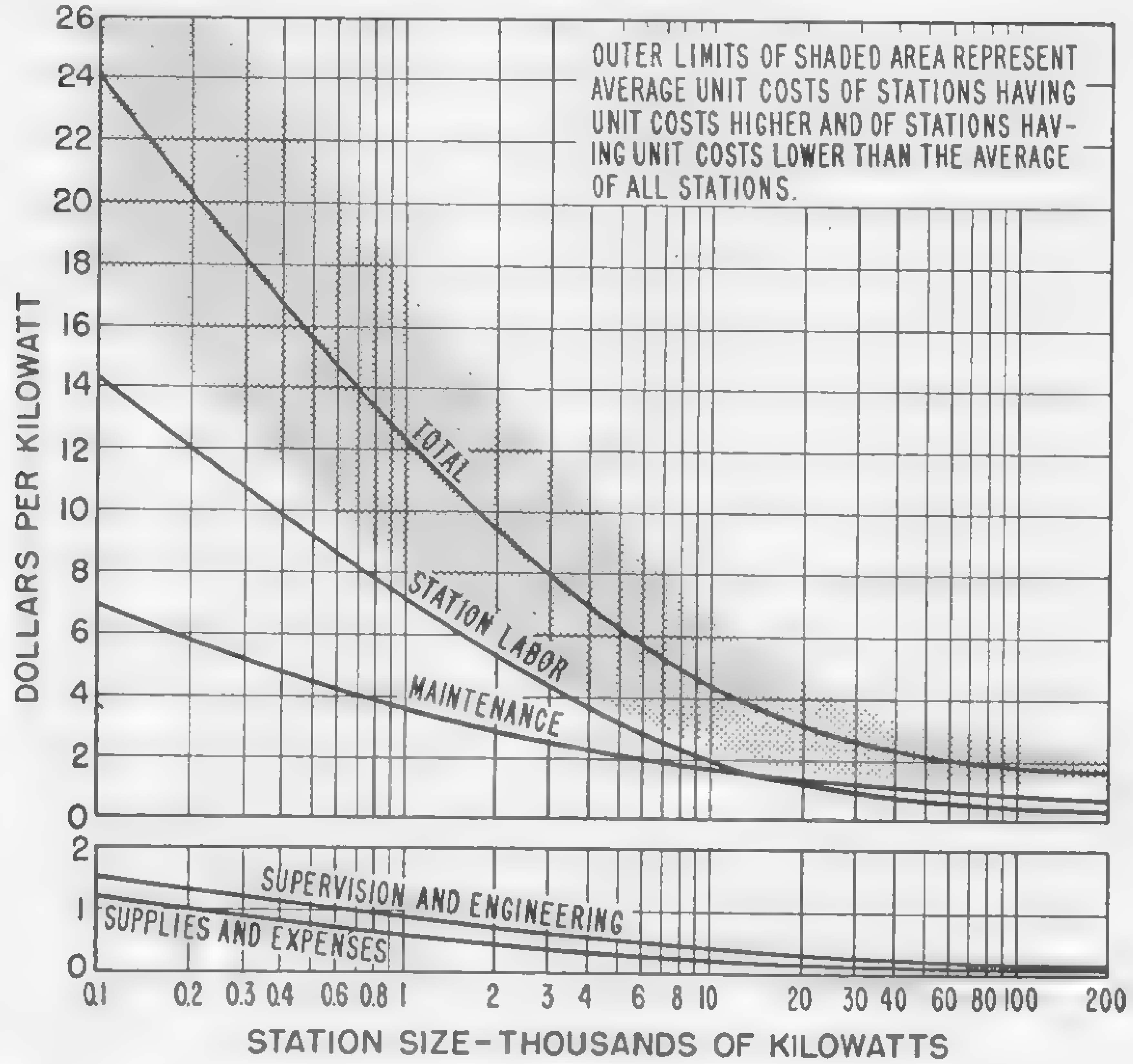


FIG. 7-2. Production expenses of a hydroelectric generating station. (Federal Power Commission)

against the hydro plant in a comparative economic analysis. However, the modern trend toward interconnection of power systems, and the necessity for placing highly efficient steam generating stations near an adequate supply of water for condensing purposes, tend to reduce the magnitude of transmission liability. In the Hoover Dam installation, for example, the transmission liability is very high for the bulk of the power produced. But the power generated is largely a by-product of the use of water for other purposes, and hence cannot properly be considered as an alternative to a steam plant placed closer to the load center. It should be reiterated here that a hydro plant serves to conserve exhaustible fuels such as coal, gas, and oil. This function lends to hydro a long-term value that cannot be readily evaluated.

The transmission plant consists of the transmission line and the transmission substations. Transmission lines are classified (1) as to type of support and (2) as single circuit or double circuit. The types of support are steel towers, wood H frames, and wood poles. The cost of transmission lines per mile is based on the voltage, and substation cost

on the capacity in kilovolt-amperes. The Federal Power Commission made a study of transmission plant costs for plants in service as of December 31, 1948, and also of plants placed in service in 1948 and 1949.* Table 7-3 gives a resume of the cost for more recent plants. The construction costs are exclusive of land, land rights, and clearing. The operation expenses are exclusive of supervision and engineering.

TABLE 7-3

AVERAGE CONSTRUCTION AND OPERATING EXPENSES OF TRANSMISSION PLANTS, 1948–49

Transmission Lines

Type of Support	Volts	Construction Cost per Mile	Annual Operating Expense per Mile of Line
Steel towers, double circuit	66,000	$22,000	$135
Steel towers, double circuit	220,000	33,000	290
Steel towers, single circuit	66,000	15,000	85
Steel towers, single circuit	220,000	26,000	180
Wood H frames, single circuit	44,000	6,500	58
Wood H frames, single circuit	154,000	11,200	105
Wood pole, single circuit	15,200	3,500	55
Wood pole, single circuit	66,000	6,500	95

Substations

Kilovolt-Amperes	Investment, dollars per kilovolt-ampere, av.	Av. Operating Expense, cents per kilovolt-ampere
5,000	17.00	37
100,000	13.50	29
1,000,000	12.00	28
2,000,000	11.50	28
4,000,000	11.00	28

Notes: Intermediate average values of construction costs may be determined by plotting a straight line between each of the above extremes on log plotting paper.

The costs of land, land rights, and clearing for transmission lines depend upon the population density and the character of the terrain. The average ratio of these costs to investment in transmission plant ranges from 5 per cent in areas where the population density is less than 25 persons per square mile to 20 per cent where the population is more than 500 persons per square mile.

Supervision and engineering expenses are expressed as a per cent of total transmission expenses. This per cent averages from 25 per cent when the total transmission expenses are $5000, 16 per cent for $100,000, to 12½ per cent for $3,000,000.

* *Electric Utility Cost Units, Transmission Plant*, Federal Power Commission (F.P.C. S-88) (Washington, D. C.: Government Printing Office, 1951).

7-9. Cost of Steam Plants, Installation. The cost per kilowatt of installed capacity for steam plants or unit of capital investment will vary widely with the size; cost of real estate; price index; plant type, whether outdoor, indoor, semi-outdoor; type of equipment installed; and reserve capacity provided. The inclusion of fuel-saving devices, automatic equipment, and high-pressure boilers reduces not only the production cost because of improved thermal efficiency, but also the

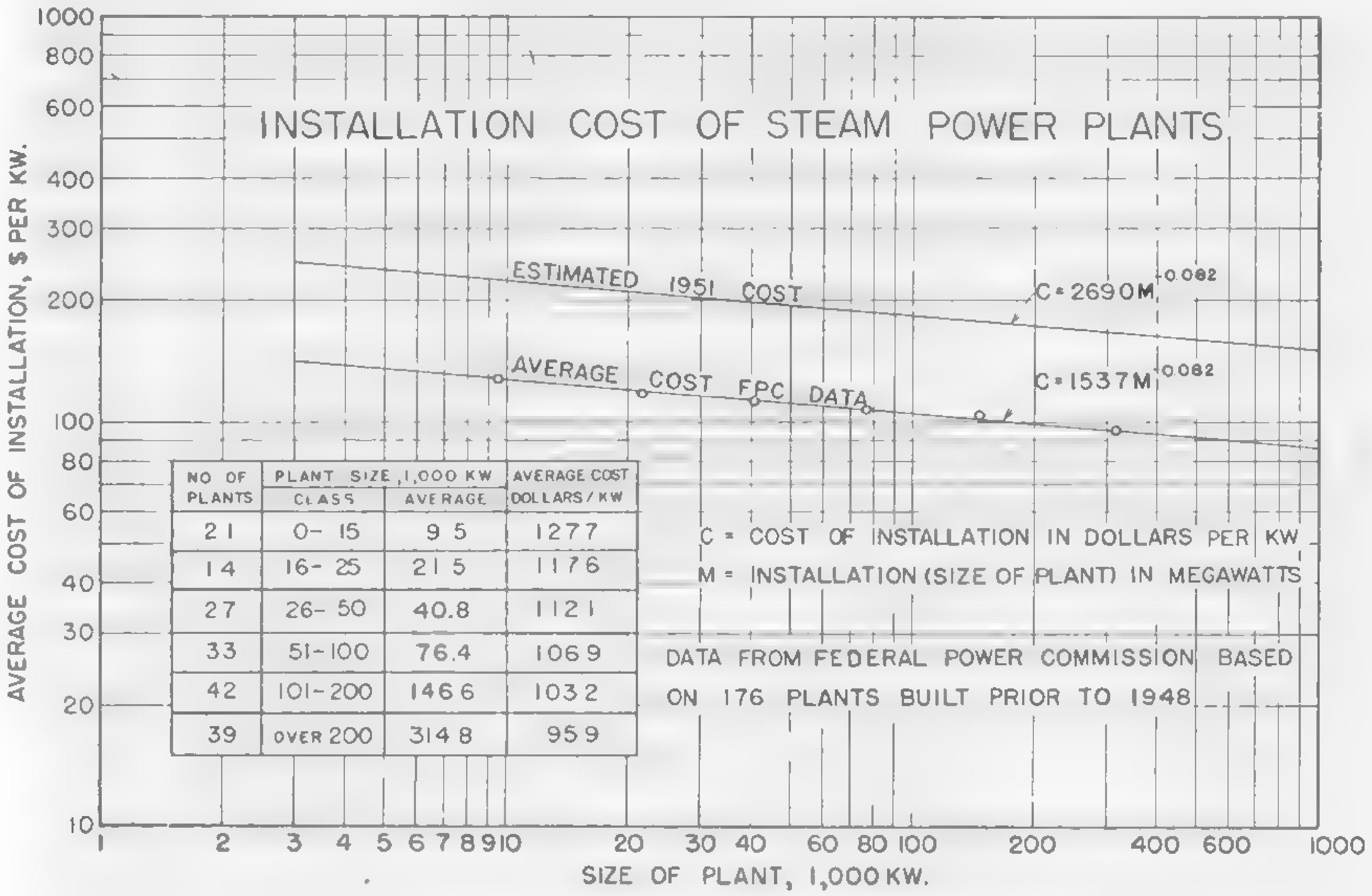


FIG. 7-3. Installation costs of steam power plants.

number of employees required to operate the plant. The 1948 report of the Federal Power Commission gives the unit installation cost per kilowatt installed for 176 plants.*

Figure 7-3 shows the average cost for different installation capacities. Some of these plants were built when the price index was lower than 1953 prices. The upper curve in Fig. 7-3 is based upon an analysis of price indexes but represents little more than a guess as to 1953 prices. The lower curve is based upon a derived equation:

$$C = 153.7M^{-0.082}$$

in which C is the cost in dollars per kilowatt installed and M is the total installation in megawatts (1000 kw). This equation was derived

* *Steam Electric Plant Construction Cost and Annual Production Expenses, 1948*, Federal Power Commission (F.P.C. S 71) (Washington, D. C.: Government Printing Office, 1948).

from the values for the extreme points shown in the table in Fig. 7–3. The calculated intermediate points fitted the line very closely. The cost of a 132,000-kw modern steam plant, completed in 1952, was 22 million dollars, or 168 dollars per kilowatt. The equation in Fig. 7–3 for a plant of this capacity indicates 179 dollars per kilowatt.

7–10. Fixed Charges, Steam Plants. Fixed charges are determined by applying a percentage rate to the capital investment. The percentage rate will vary with the money market, value of real estate, and life of the installed equipment. Total fixed charges are made up of: (1) Cost of money (which includes interest on debt), cost of floating the loan, levying and collection of taxes (public enterprises); (2) taxes and insurance; and (3) depreciation. A general statement in regard to the rates which may be applied to each of these items is shown in Table 7–4.

TABLE 7–4

Item	Rates (%)
Cost of money or interest	6.5
Taxes and insurance	2.5
Amortization or depreciation and obsolescence	4.5
Total, fixed charges	13.5

7–11. Production Costs, Steam Plants. Production or operating costs for steam electric plants form a large part of the total cost of generation of electric power. The production costs include the cost of fuel, labor, and maintenance. The unit cost of fuel depends upon the price paid for fuel—coal, oil, or gas—and the thermal efficiency of the plant. Coal is purchased by the ton, oil by the gallon or 42-gallon barrel, and gas by the cubic foot. However, the engineer is interested in the number of British thermal units (Btu) contained in the fuel and therefore reduces the cost to cents per million Btu. The calorific power in Btu for coal ranges from 11,000 to 16,000 per lb; oil from 142,000 to 155,000 Btu per gal; and gas from 900 to 1,200 Btu per cu ft. Figure 7–4 is based upon a fuel cost of 20 cents per million Btu. This cost would correspond to 12,000 Btu per lb coal at $4.80 per ton, 150,000 Btu per gallon of fuel oil at $1.25 per 42-gal barrel or 1,000 Btu per cu ft gas at 20 cents per 1,000 cu ft. A study of average fuel costs (1948) in cents per million Btu reveals the following: *

* *Ibid.*

No. Plants	Fuel	Cost in Cents per Million Btu		
		Average	Min.	Max.
152	Coal	28.23	13.43	47.67
35	Oil	41.79	16.40	65.00
57	Gas	11.06	3.30	28.56

The cost of fuel varies with the geographic location of the source of the fuel (transportation charges) and the quality of the fuel in Btu per unit. The fuel cost per kilowatt-hour of net generation depends upon

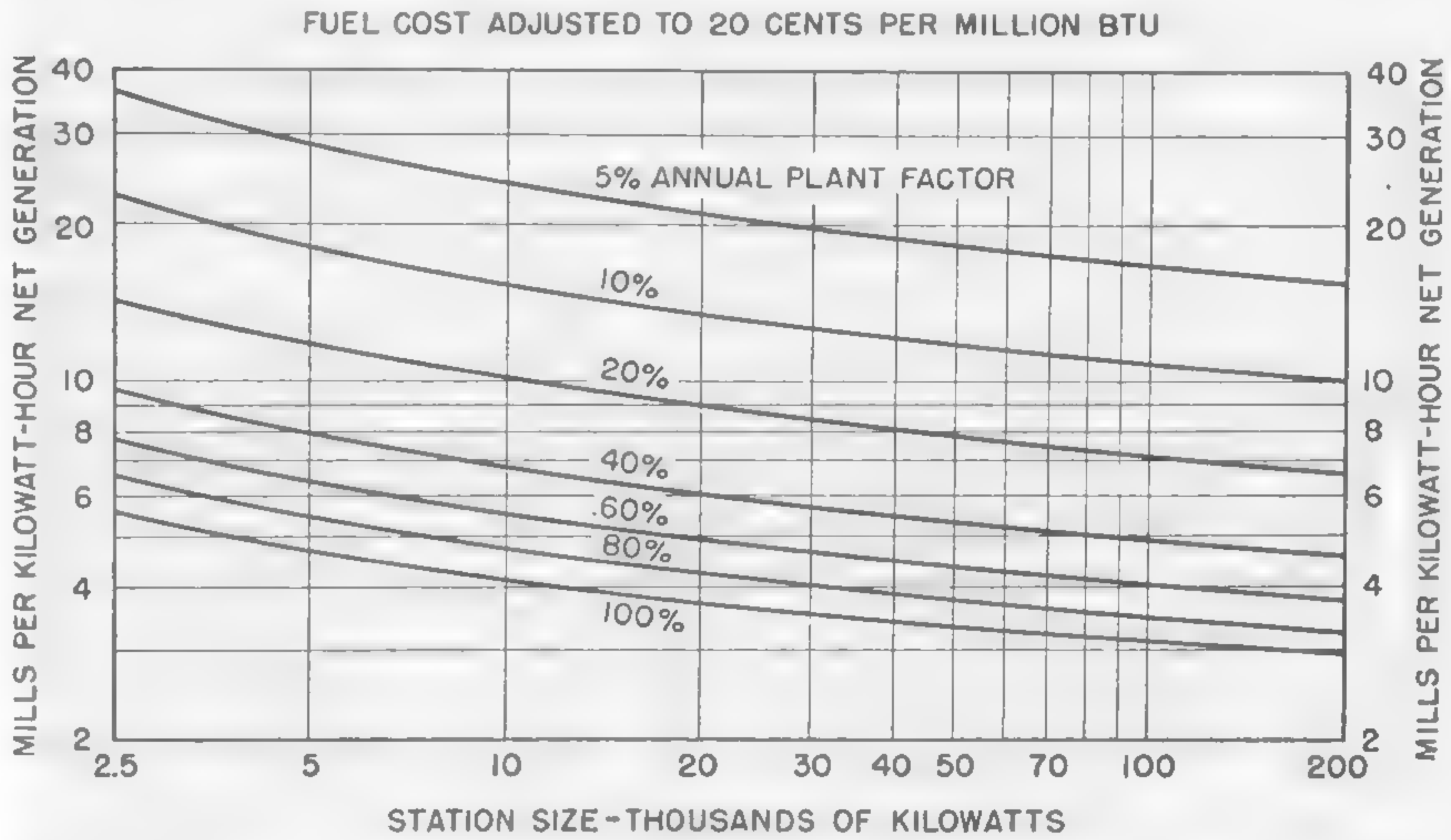


FIG. 7–4. Production expenses of a steam-electric generating station. (Federal Power Commission, *30th Annual Report*, 1950, p. 33)

the cost per million Btu and the thermal efficiency of the equipment and the load factor. Based on a study of 217 plants, the range in average Btu per kilowatt-hour of net generation ranges from 10,588 to 25,481. The fuel cost in mills per kilowatt-hour ranges from 1.58 mills to 11.22 mills for 152 coal-burning plants, 1.29 mills to 9.31 mills for oil-burning plants, and from 0.47 mill to 4.75 mills for 57 gas-burning plants.

The production cost exclusive of fuel, including (1) operation labor, supervision, and engineering; (2) water; (3) operation supplies and expenses, and other miscellaneous expense averages 1.12 mills per kwh with a range of from 0.27 minimum to 3.18 maximum. The median value is 1.07 mills per kwh. The total number of plants studied is 207.

Other items such as supervision, supplies and expenses, maintenance, and station labor are also included in the production cost for steam plants.

Figure 7–4 shows average production costs (exclusive of fixed charges) for privately owned electric utilities. The fuel cost is adjusted to 20 cents per million Btu.

7–12. Hydro Versus Steam. At one time the economic justification of a hydro plant was based upon the comparative cost of the development of a similar block of power through the construction of a steam plant. Such a procedure may be a valid criterion when an alternative project deals solely with the development of power. However, with

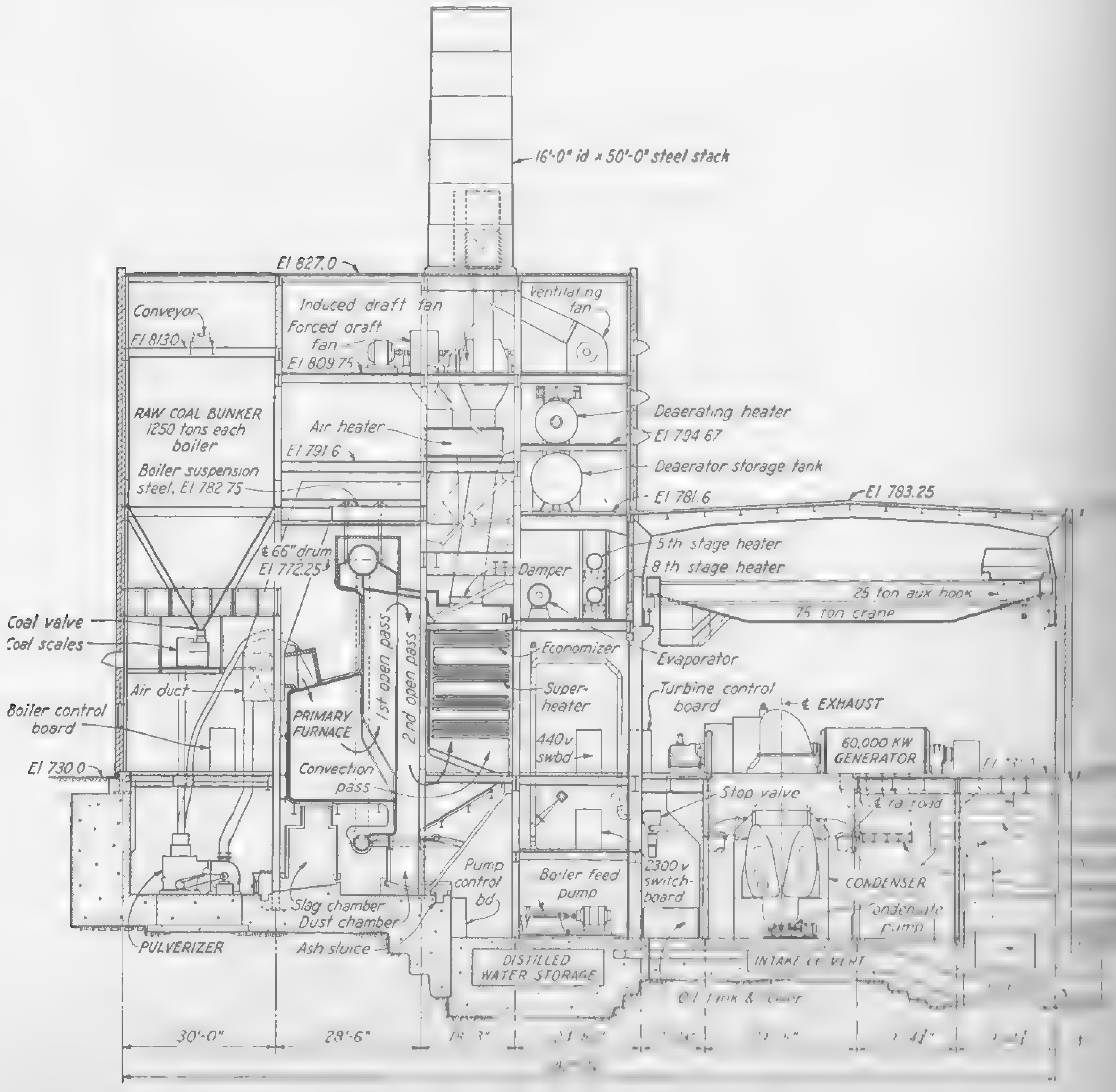


Fig. 7–5. Section of Watts Bar steam plant. (Tennessee Valley Authority)

the rapid growth of multiple-purpose developments, the appearance of a power component often becomes automatic and must be studied for its own economic justification without regard to its counterpart in steam. It should also be remembered that there is no loss of expendable natural resources involved in hydro power developments and for that reason it has an advantage over steam.

Steam has an advantage over hydro in the matter of plant location. Hydro must be developed at the site where it is found and frequently involves long and costly transmission lines to carry power to the load center. Steam plants, on the other hand, may be located closer to the load center. Modern steam plants must, however, be placed close to a source of a water supply for furnishing the necessary cooling water for condensing purposes. This requirement averages from 40 to 80 gal per kwh, depending upon the temperature of the cooling water. Figure 7–5 shows a section of the Watts Bar Steam Plant which operates in conjunction with the hydro power plant system of the Tennessee Valley Authority.*

* See "Economic Aspects of Energy Generation," *Trans. A.S.C.E.*, Vol. 104 (1939), 942; and "Cost of Energy Generation," *Trans. A.S.C.E.*, Vol. 104 (1939), 1050. Both of these papers are excellent, but it should be remembered that the prices of labor and materials have increased considerably since the time of writing and due allowance must be made for the values developed in the papers.

CHAPTER 8

HYDRO-STEAM ASSOCIATION

8–1. Introduction. When a hydro plant serves as the sole source of power to carry a given electric load, the maximum available firm power is limited by the minimum water supply. Hydro plants perform much more economically and with greater flexibility of operation when they are interconnected with other power sources. The combination of hydro generation of electric energy with that of steam, when both installations are economically feasible, usually provides cheaper power than either one operating independently. When operated in conjunction with fuel power plants, hydro plants may carry all or part of the base load up to the installed capacity during periods of high stream flow and may carry the peak load during the period of low stream flow. The amount of the load carried by the fuel plant may therefore be adjusted to conform with the available water supply and hence with the power potential of the hydro plant. This plan of operation tends toward a greater utilization factor (ratio of the amount of energy developed to the amount of energy available in the stream) of the available water supply, and hence a saving in the amount of fuel consumed at the steam plant. The amount of the savings will be reduced by the annual cost of the fixed charges and electric losses on the transmission line if the hydro plant is located at a distance from the load center. In the case of Niagara power, for instance, even when the stream flow is fairly constant, better economy results from a tie-in with steam power, because no load curve requires a constant output and the steam power can be employed to carry peak loads which exceed the base load available from hydro.

It has already been stated that in the case of the Tennessee Valley Authority the installed steam capacity is expected to exceed the installed hydro capacity in 1954. The Safe Harbor and Holtwood hydro plants on the Susquehanna River are combined with steam plants to carry the electric load of Washington, D. C., and Baltimore, Md. Table 8–1 shows the results of studies made to determine savings that resulted from the development of the Conowingo hydro plant in serving the Philadelphia market. The following data on the cost of energy generation are quoted from a paper by Ezra B. Whitman in Table 8–1.

TABLE 8–1

Cost of producing all power requirements from steam (1930)		$12,666,500
Cost of steam power after hydro power comes in (1930)		7,163,000
Operating saving from hydroelectric energy		$ 5,503,500
Fixed charges on $20,000,000 representing 150,000 kw saved in steam-plant investment		2,330,000
Charges on saving in coal and equipment supplies from use of hydro-electricity		82,000
Total representing what the Company could afford to pay for the hydro plant		$ 7,915,500
Cost of Hydro Power:		
Rentals	$4,580,000	
Operating expenses, taxes, and depreciation on hydro plant and lines	1,430,000	
Operating expenses, taxes, and depreciation on investment in Company territory in relation to hydroelectric generation	1,126,000	$ 7,136,000
Remainder representing the saving from use of hydro plant		$ 779,500

Whitman continues: "As the hydro plant was estimated to furnish 1,150,000,000 kw-hr per yr, the saving on this part of the current generated at the Conowingo hydro plant would amount to 0.7 mill per kw-hr or, on the total power generated of 2,230,000,000, the saving would be approximately 0.36 mill. The total cost of the water-power current delivered in Philadelphia was estimated at 6.2 mills per kw-hr. The total cost of the steam generation of all the current was estimated to cost 5.68 mills per kw-hr; but by utilizing the hydro-power for peak service during the dry flow and base-load service during the wet months, the cost of the combined energy generation was reduced nearly $800,000 per yr, although the hydro-power by itself cost 6.2 mills as compared with the 5.68 mills if all the power had been generated by steam alone. This is a typical illustration of the economic advantage of combining steam and hydro-electric generation." *

8–2. Load Curve. As previously stated, the load curve presents data on the amount of power required to be delivered for each instant of a day, week, or month. The area under the load curve equals the total power output in kilowatt-hours. The weekly load curve will present all of the varying characteristics of a typical utility load. The selection or assumption of a weekly load during December will ordinarily indicate the maximum weekly load demand. In studying hydro-steam association it is sometimes convenient to transform the chronological load curve into a power-per cent of time curve. This is done by rearranging

* Ezra B. Whitman, "Cost of Energy Generation," *Trans. A.S.C.E.*, Vol. 104 (1939), 1115.

the hourly kilowatt demands in a decreasing order of magnitude and calculating the per cent of time or total hours for which each demand is sustained. The power-per cent of time curve is then plotted using power as ordinates and time as abscissas. An auxiliary scale of ordinates is supplied to indicate the per cent of the total load area which lies above corresponding horizontal lines.

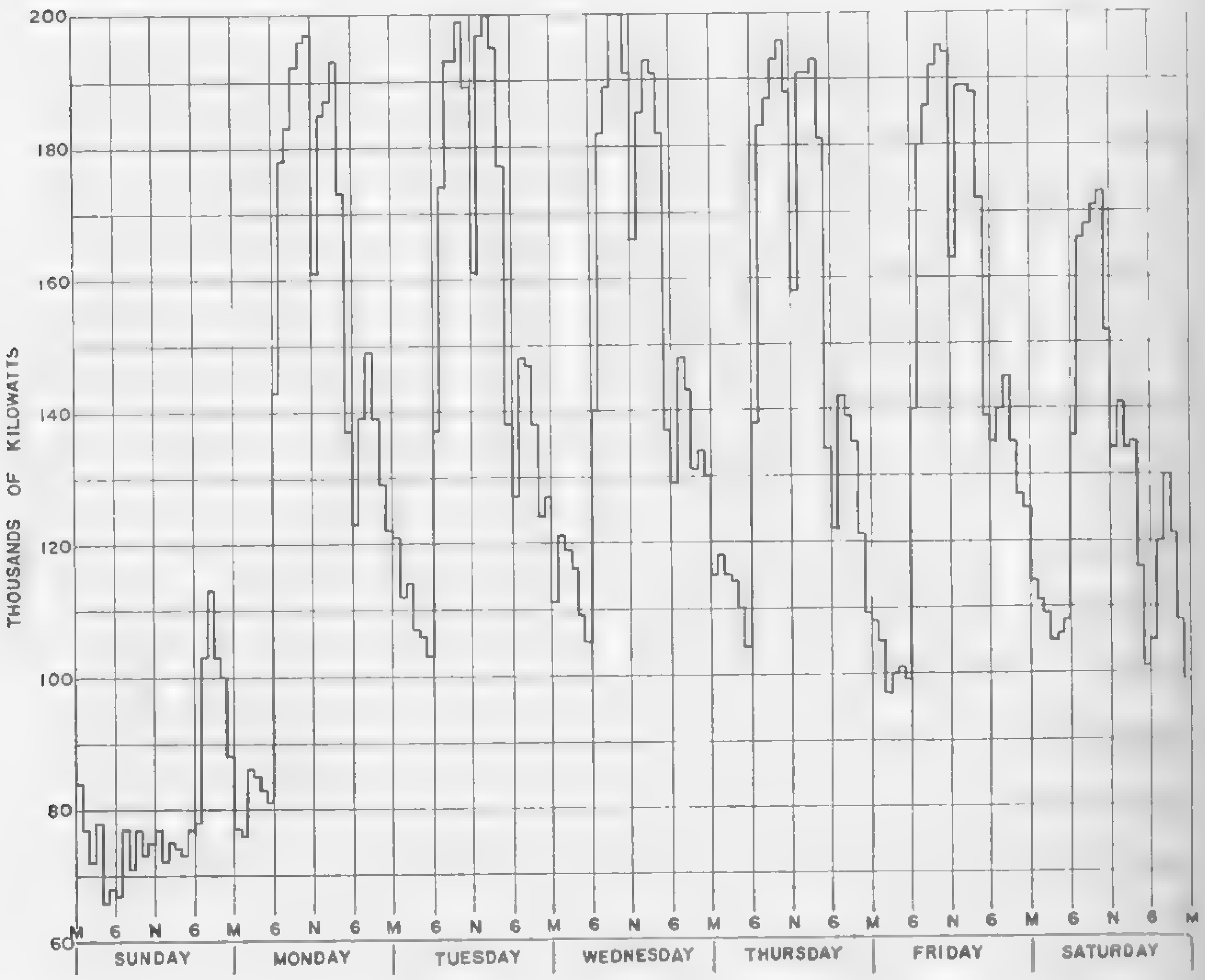


Fig. 8-1. Weekly load curve for December.

An example is presented here to illustrate the procedure. For convenience in computation, the instantaneous load curve is modified by considering the average hourly demand as continuous for one hour. This procedure results in indicating a peak load lower than the instantaneous peak but the error is relatively unimportant. Assume that the data in Table 8-2 give chronological information for a December weekly load curve. These data are plotted in Fig. 8-1. The power distribution curve is shown in Fig. 8-2. The total area under the curve is 22.795 million kilowatt-hours. The auxiliary scale on the right side of Fig. 8-2 shows the per cent of total area which lies above the indicated horizontal line.

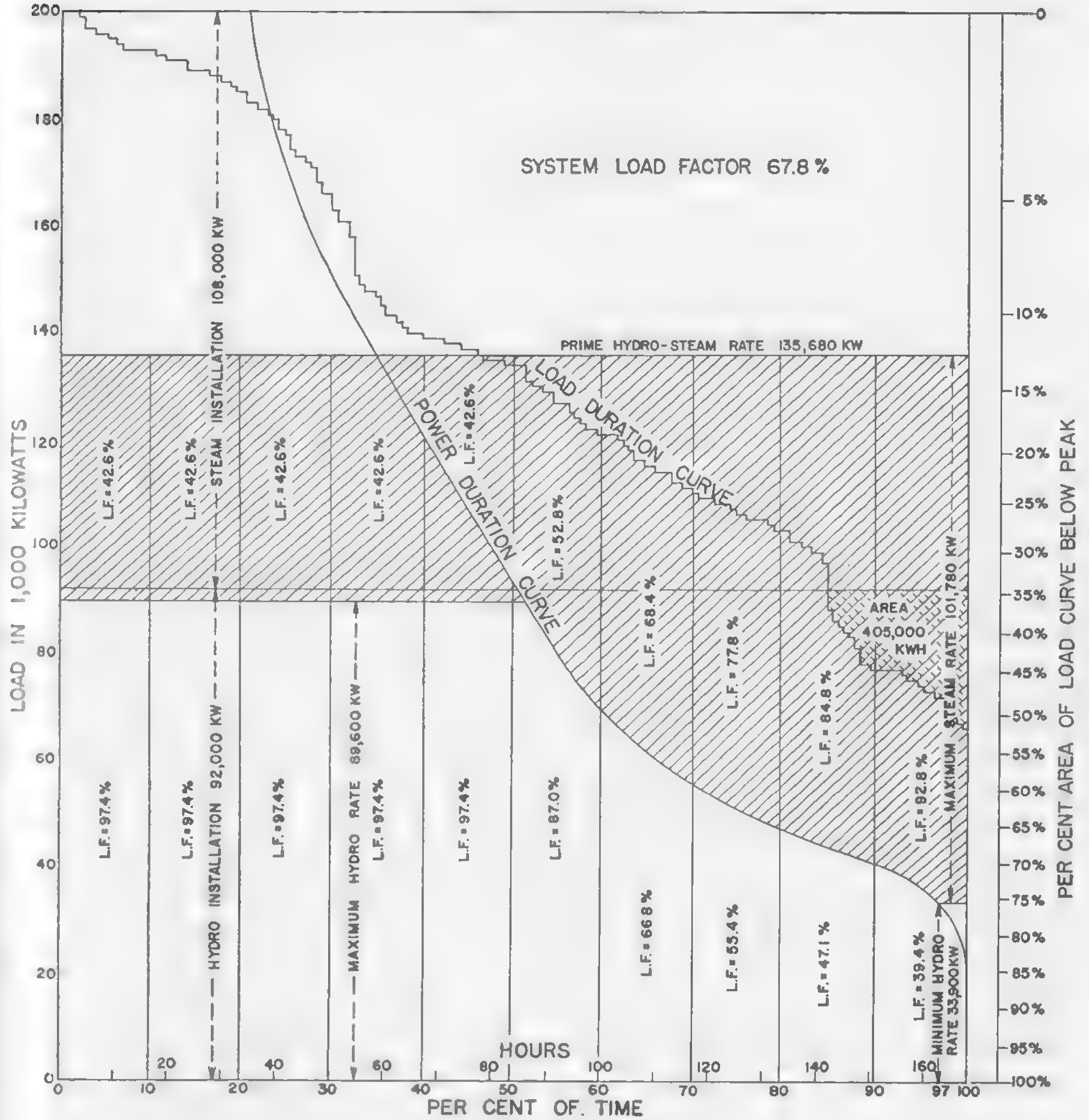


Fig. 8-2. Combined hydro-steam study.

8-3. Hydro Power. The analysis of the performance of a hydro plant which will operate on the given load curve requires the determination of the available power from the hydro plant. Let us assume the following: the available water supply is as shown in Fig. 8-3, the average net effective head is 65 ft, the efficiency is 88 per cent, and the power factor is 1.00. The power site lends itself to the development of adequate pondage facilities. Data from Fig. 8-3 has been transferred to Fig. 8-2 and is shown as a power duration curve.

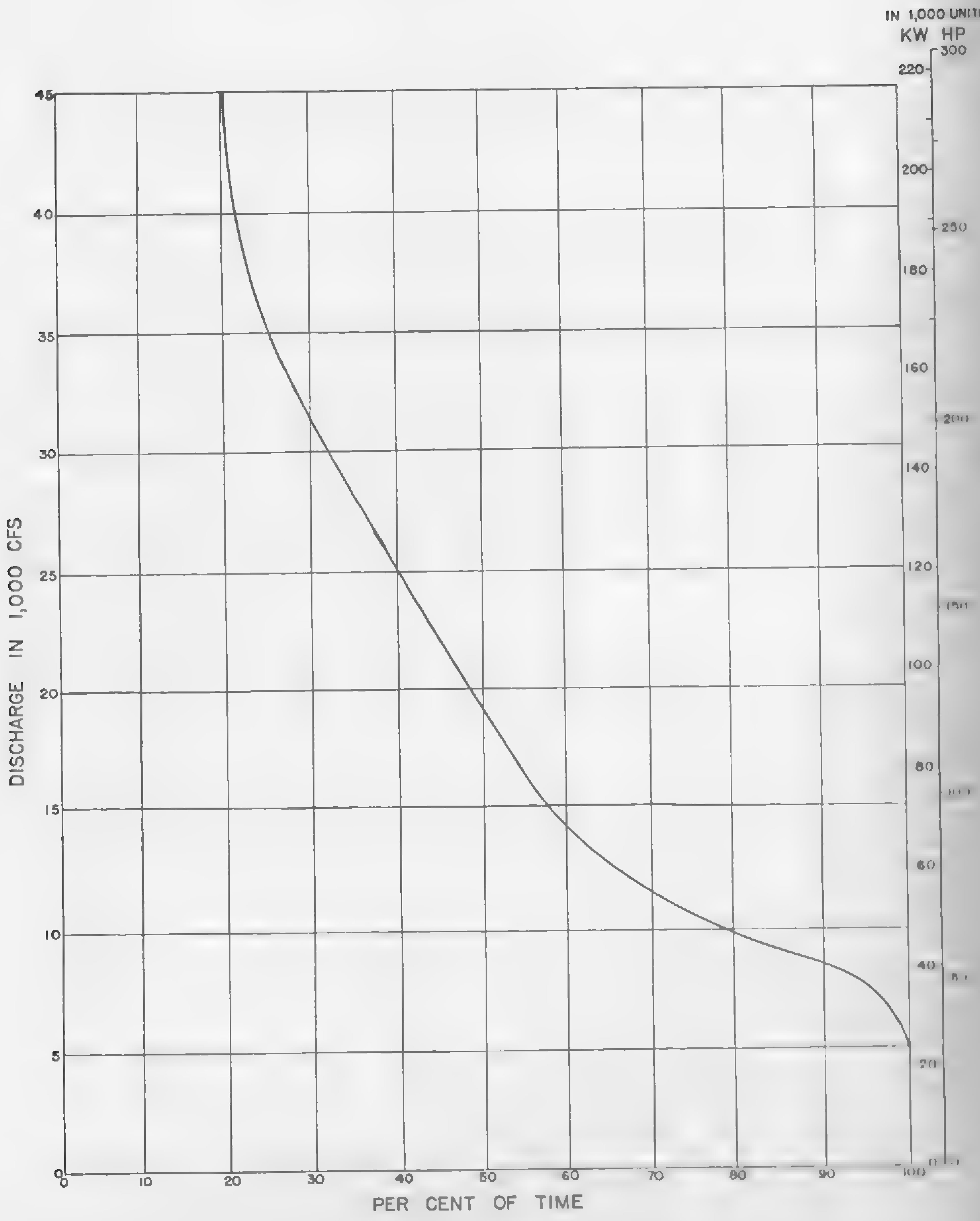


FIG. 8-3. Flow-duration curve.

TABLE 8-2

LOAD IN 1,000 KW

Time	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
1 A.M.	84	77	121	111	115	108	114
2	77	76	112	121	118	105	111
3	72	86	114	119	115	97	109
4	78	85	107	116	114	100	105
5	66	83	106	109	110	101	106
6	68	81	103	105	104	99	108
7	67	143	137	140	138	140	136
8	77	178	174	182	183	180	166
9	71	183	193	189	187	186	168
10	77	192	193	200	193	192	171
11	73	196	199	200	196	195	173
12	75	197	189	191	188	194	152
1 P.M.	77	161	161	166	158	163	134
2	72	185	197	185	191	189	141
3	75	187	200	193	191	189	134
4	74	193	195	191	193	188	135
5	73	173	177	182	181	172	116
6	77	137	138	137	134	139	101
7	78	123	127	129	122	135	105
8	103	139	148	148	142	140	120
9	113	149	147	143	139	145	130
10	103	139	138	131	135	135	121
11	100	129	124	134	121	127	108
12	88	122	127	130	109	125	99

Total: 22,795,000 kwh
Minimum: 66,000
Maximum: 200,000

Consider the flow at 97 per cent of the time as 7000 cfs, then the firm power which can be produced is:

$$\text{kw} = \frac{7000 \times 62.5 \times 65 \times 0.88}{738} = 33{,}910 \text{ kw}$$

or

$$\text{hp} = \frac{7000 \times 62.5 \times 65 \times 0.88}{550} = 45{,}500 \text{ hp}$$

During a period of low flow the hydro plant could produce in one week:

$$33{,}910 \times 168 = 5{,}696{,}880 \text{ kwh}$$

If the low flow occurs during a maximum load week and the hydro were operated on the peak, the hydro plant would develop 25 per cent of the total load (5,696,880 = approx. 25 per cent of 22,795,000).

Inspection of the load duration curve (Fig. 8–2) indicates that 25 per cent of the area of the load curve lies above the horizontal line designating 108,000 kw, or 92,000 kw below the peak load of 200,000 kw. Therefore the hydro installation should be at least 92,000 kw. This value would have to be increased to offset transmission losses between the hydro plant and the distribution point at the load center. The steam installation required would be the difference between 200,000 kw and 92,000 kw, or 108,000 kw.

8–4. Economic Justification. In order to determine the economic feasibility of the operation of the hydro plant in connection with the given load, it is necessary to estimate the production cost of both the hydro and the steam in terms of mills per kilowatt-hour. Let us assume the following:

Hydro

Installation: 92,000 kw
Cost of installation: $350 per kw (Fig. 7–1)
Production expenses (operation and maintenance) at 100 per cent load factor: $1.92 per kw (Fig. 7–2)
Fixed charges: 9.5 per cent

Steam

Installation: 108,000 kw
Cost of installation: $182 per kw (Fig. 7–3)
Fuel: 12,000 Btu per lb of coal
Cost: $4.80 per ton = $0.20 per million Btu
Performance: 15,000 Btu per kwh
Operation and maintenance: See Fig. 7–4 and Table 8–2
Fixed charges: 13.5 per cent

The unit cost of production, based on these assumptions for various load factors, is shown in Table 8–2.

The unit production cost of hydro is determined by adding the fixed charges ($0.095 \times 350 = \$33.25$) and the operation and maintenance ($1.92), and dividing the sum ($35.17) by 8760 times load factor.

The total unit cost of steam production in mills per kilowatt-hour is determined by adding the fixed charges ($0.135 \times 182 = \$24.57$ or 24,570 mills) divided by 8760 times the load factor to the production cost obtained from Fig. 7–4.

8–5. System Load Factor. The system load factor for combined hydro-steam is

$$\frac{22,795,000}{168 \times 200,000} = 67.8 \text{ per cent}$$

The prime hydro-steam rate is therefore

$$0.678 \times 200,000 = 135,600 \text{ kw}$$

8–6. Hydro Load Factors. It is assumed here that the entire available water supply is used for developing power up to the installation capacity of 92,000 kw and within the load demand. It will be noted in Fig. 8–2 that the lower part of the load duration curve falls below the 92,000-kw installation. The area lying above the load curve and below the installation line is 405,000 kwh. The total weekly production of the hydro plant during a period of high flow would then be

$$92,000 \times 168 - 405,000 = 15,051,000 \text{ kwh}$$

and the maximum hydro rate would be

$$\frac{15,051,000}{168} = 89,600 \text{ kw}$$

and the maximum hydro load factor is

$$\frac{89,600}{92,000} = 97.4 \text{ per cent}$$

This load factor could continue until the flow drops below the possibility of developing the capacity of the hydro plant, slightly beyond 50 per cent of the time in Fig. 8–2. As the flow continues to reduce, the load factor may be determined by dividing the average ordinate to the load duration curve by the hydro installation (Table 8–3).

TABLE 8–3

COST OF PRODUCTION, IN MILLS PER KWH

	Hydro	Steam		
Load Factor L.F.	Unit Cost $\frac{35,170}{8760 \times \text{L.F.}}$	Fixed Charges + 13.5% × 182 ÷ L.F. × 8760	Production or Operation and Maintenance	Total Unit Cost
1.00	4.0	2.81	3.10	5.91
0.90	4.5	3.12	3.30	6.42
.80	5.0	3.51	3.50	7.01
.70	5.7	4.01	3.75	7.76
.60	6.7	4.68	4.00	8.68
.50	8.0	5.61	4.40	10.01
.40	10.0	7.02	4.80	11.82
.30	13.4	9.35	5.90	15.25
.20	20.1	14.03	7.00	21.03
.10	40.8	28.1	11.00	39.10

8–7. Steam Load Factors. The hatched area in Fig. 8–2 represents the amount of power in kilowatt-hours that must be produced by steam. As long as the water supply is capable of producing power up to the capacity of the hydro plant, the lower boundary of this area is the horizontal line representing the maximum hydro rate. The flow duration curve becomes the lower boundary after it is intersected by the maximum rate. The upper boundary is the horizontal line representing the hydro-steam rate. Load factors for the steam installation for different per cents of time are obtained by dividing the measured length of average ordinates between the horizontal line and the hydro line by the steam installation (Table 8–4).

TABLE 8–4

Per Cent of Time	Load Factors	
	Hydro	Steam
0– 10	97.4	42.6
10– 20	97.4	42.6
20– 30	97.4	42.6
30– 40	97.4	42.6
40– 50	97.4	42.6
50– 60	87.0	52.8
60– 70	66.8	68.4
70– 80	55.4	77.8
80– 90	47.1	84.8
90–100	39.4	92.8
Average	78.3	58.9

8–8. Comparative Costs. It will be noted from Fig. 8–2 and the analysis in Art. 8–7 that the total power produced by the system based on a maximum load week is $135{,}680 \times 168 = 22.8$ million kilowatt-hours. If the maximum weekly load is considered as the average weekly load, the hydro plant would produce 53 per cent and the steam would produce 47 per cent of the total load. Since the average load factor for hydro is 78.3 per cent, the average cost of production is 5.13 mills per kwh (Fig. 8–4). The average load factor for steam is 59.8 per cent, the average cost of production is 8.70 mills per kwh. The over-all cost for hydro-steam is:

$$5.13 \times 0.53 + 8.70 \times 0.47 = 6.81 \text{ mills per kwh}$$

The cost of steam only for a load factor of 67.8 per cent is 7.96 mills per kwh. Based on assumed conditions the development of the hydro plant is economically justified.

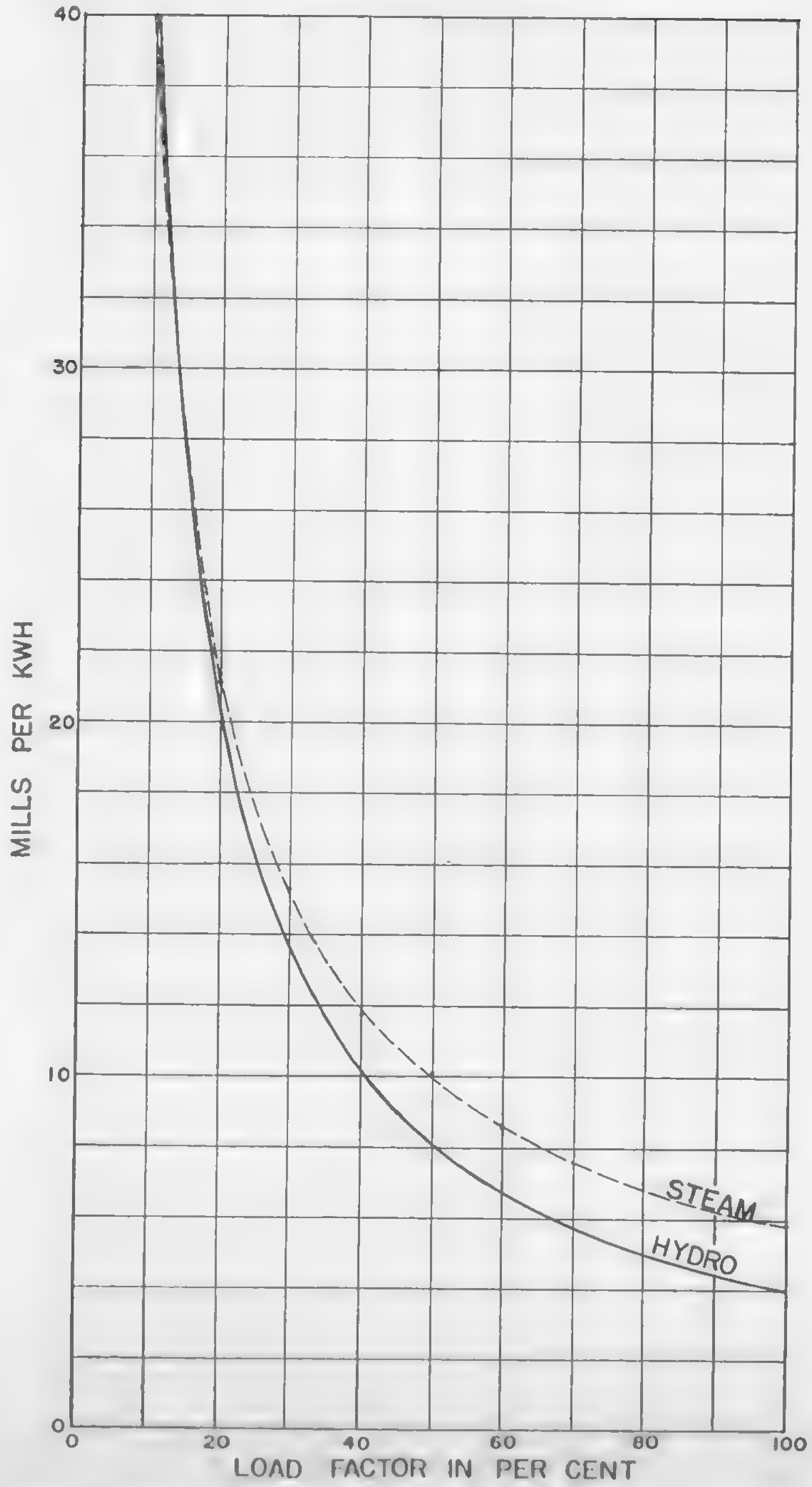


FIG. 8–4. Hydro and steam production costs, based on illustrative data.

TABLE 8–5

COST ANALYSIS AS SEPARATE SYSTEMS

	Hydro	Remarks	Steam	Remarks
Installed capacity	50,000		150,000	
Installation cost per kw	$370	Fig. 7–1	$178	Fig. 7–3
Load factor	67.8%		67.8%	
Fixed charges	35.15	at 9½%	24.03	at 13½%
Fixed operating cost per kw	2.40	Interpolated from Art. 7–7	——	
Total fixed cost, Annual	37.55		24.00	
Unit fixed cost, mills per kwh	6.33	Total fixed cost ÷ (8760 × L.F.)	4.01	Total fixed cost ÷ (8760 × L.F.)
Production cost, mills per kwh	——		3.75	
Total unit cost, mills per kwh	6.33		7.79	

Weighted combined cost as separate systems, 7.43 mills per kwh
Combined cost, associated, 6.81 mills per kwh

8–9. Discussion of Method. The above method of analysis would necessarily be subject to a number of refinements in order to arrive at a precise result. For instance, the maximum load is not the average load, but consideration of the maximum is necessary to arrive at the proper hydro installation. The assumed cost data may require considerable revision. Moreover, the annual cost of transmission lines and losses due to long distance transmission would be chargeable against the hydro installation. Some of this would be offset by the usual installation of reserve power installed in the steam plant and by the cost of fuel required to keep boilers hot and prepared to furnish power for power production when the load demanded it. The method is presented here to illustrate the basic principles involved in the study.

8–10. Advantages of Association. The previous discussion demonstrated by a specific example that the construction of the combined hydro-steam installation could produce cheaper power than a steam installation alone. A further academic illustration will show that if it is assumed that the same load were divided between independent and noninterconnected steam and hydro companies, the over-all basic cost would be increased. The hydro company operating alone would be limited by low-stream flow conditions to a firm output of 33,900 kw. During the critical week it could produce $33{,}900 \times 168 = 5{,}695{,}200$ kwh, or approximately 25 per cent of the total load of 22,795,000 kwh. It will be assumed that sufficient pondage is available to permit a peak load of 50,000 kw, which is also 25 per cent of the 200,000-kw total

peak. The hydro installation will therefore be assumed to be 50,000 kw and the steam installation 150,000 kw (no reserve capacity included). Hydro-steam association versus separate systems shows a saving of 0.62 mill per kwh. These figures are not conclusive because of several factors which are not considered. Among them are reserve capacity and transmission liability. This analysis is shown in Table 8–5.

PROBLEMS

PROBLEMS

CHAPTER 1

1–1. Prepare a brief report on U. S. Geological Survey Circular No. 329, entitled *Developed and Potential Water Power of the United States and Other Countries of the World,* by B. E. Jones and L. L. Young, 1954.

1–2. From the current report of the Federal Power Commission entitled *Production of Electric Energy, Capacity of Generating Plants 19___* prepare a report on the progress of the electric energy industry up to the last date for which Federal Power Commission Report is available.

1–3. From the data source given in Problem 1–2, (a) compute the ratio between privately owned and publicly owned plants of the total production, and the hydro, steam, and internal combustion production in kilowatt-hours; (b) compute the same ratios for total installed capacity; (c) plot curves which will indicate changes in the trends since 1937.

CHAPTER 2

2–1. The average annual yield of the Columbia River at Grand Coulee Dam for 1913–44 was 75,116,000 acre-ft. A dam was built to provide an average head of 300 ft. Estimate the potential energy in kilowatt-hours available from this installation. Compare your answer with the actual plant installed. See Table 5–1 and Eq. (2–1).

2–2. The design head at Grand Coulee power plant is 330 ft, the discharge is 4660 cfs, and the horsepower is 154,000. Estimate the efficiency and determine the capacity in kilowatts.

2–3. Using Eq. (2–5), prepare a curve showing the variation of the ratio of discharge diameter to nominal diameter against the specific speed for reaction turbine runners. At what specific speed do the two diameters become identical?

2–4. Prove that the specific diameter of a turbine runner is equal to the ratio between the unit speed and the specific speed, or

$$D_s = \frac{N_1}{N_s}$$

2–5. The test of a 16-in. runner under a 25-ft head gave the following as the best results:

Speed = 400 rpm
Discharge = 17.5 cfs
Output = 39.8 hp

Required:

a) Find the turbine characteristics: ϕ, N_s, N_1, P_1, Q_1, and the efficiency.

b) What type of wheel is this?

c) Suppose that a 40-in. runner of the same design is used under a 150-ft head. Compute the speed, discharge, and horsepower.

d) Suppose that turbines of the type which you have identified were satisfactory for a certain plant but that the number of units (and consequently the power of each) and the speed had not been decided upon. If the capacity of the plant were 25,000 hp and head is 150 ft determine respectively the speed and size of wheel for 2-, 4-, and 5-unit combinations.

2–6. The test of an 85-in. runner under a 47-ft head gave the following as the best results:

Speed = 200 rpm
Discharge = 865 cfs
Output = 3450 hp

Required:

a) Find the turbine characteristics: ϕ, N_s, N_1, P_1, Q_1, and the efficiency.

b) What type of wheel is this?

c) Suppose that a 50-in. runner of the same design is used under a 100-ft head. Compute the speed, discharge, and horsepower.

d) Suppose that turbines of the type which you have identified were satisfactory for a certain plant but that the number of units (and consequently the power of each) and the speed had not been decided upon. If the capacity of the plant were 30,000 hp and head is 80 ft, determine respectively the speed and size of wheel for 2-, 4-, and 6-unit combinations.

2–7. A model Francis turbine having a nominal diameter of 5 in. was tested. The result indicates a turbine efficiency equal to 95.5 per cent. Predict by the Moody formula the efficiency of the prototype runner having a nominal diameter of 12.5 ft.

2–8. The diameter D_1 of the Unit 3 turbine at Shasta Dam is 171 in.; the design head is 380 ft; the power P at the design head is 132,000 hp; the discharge is 3,500 cfs; and the speed is 138.5 rpm. Compute the turbine constants, ϕ, N_1, Q_1, P_1, N_s, D_s, and m for this turbine.

2–9. How many poles should be in the generator for the Shasta turbine if the frequency of the current is 60 cycles per second?

2–10. The area of the 12.5-ft diameter horseshoe tunnel shown in plan in Fig. 2–10, page 35, for the Ocoee No. 3 installation is 129 sq ft. The tunnel serves one unit in the power plant. The turbine develops 33,500 hp at 272 ft with an efficiency of 90.4 per cent. What is the average velocity in the tunnel?

2–11. The Tennessee Valley Authority is developing a pumped storage hydro plant at Hiwassee Dam. The installed machine will operate as a turbine during peak load periods and as a pump during off load periods. Prepare a brief report on this project from data given in *Civil Engineering*, March, 1953, pp. 56–59. See Figs. 1–3 and 1–4 for location and profile.

CHAPTER 3

3–1. The following table gives the mean weekly natural discharge at a potential power site at which an average head of 40 ft can be developed. The record contains the three driest years within an over-all period of record of 32 years.

Natural Flow by Weeks

[Mean weekly discharge in cubic feet per second]

Week Ending	Week	1921	1922	1923	1924	1925	1926	1927	1928	1929
Jan. 7..	1	90,800	67,900	61,000	173,000	67,800	28,900	352,000	128,000	28,300
14..	2	64,800	57,400	61,800	222,000	100,000	45,100	233,000	72,400	51,000
21..	3	126,000	91,900	45,500	183,000	136,000	52,200	57,400	62,900	68,400
28..	4	82,400	198,000	112,000	142,000	98,500	160,000	110,000	63,200	133,000
Feb. 4..	5	75,900	189,000	198,000	71,200	60,000	85,700	121,000	65,300	179,000
11..	6	127,000	76,600	254,000	51,600	51,000	87,800	145,000	68,200	69,200
18..	7	244,000	88,400	257,000	448,800	61,100	61,000	131,000	82,400	71,200
25..	8	203,000	188,000	184,000	93,700	120,000	61,200	151,000	54,600	77,500
Mar. 4..	9	119,000	176,000	72,900	* 194,000	83,800	79,300	187,000	* 48,600	188,000
11..	10	85,500	269,000	104,000	182,000	48,300	73,200	180,000	70,100	236,000
18..	11	98,600	313,000	217,000	141,000	44,500	90,400	278,000	134,000	252,000
25..	12	60,600	274,000	239,000	78,600	53,500	67,700	232,000	120,000	259,000
Apr. 1..	13	85,100	120,000	192,000	81,400	48,600	59,400	107,000	81,600	305,000
8..	14	97,000	170,000	99,600	79,600	36,800	64,100	114,000	99,400	258,000
15..	15	64,500	128,000	95,200	84,200	36,300	61,600	188,000	82,600	114,000
22..	16	178,000	103,000	119,000	101,000	33,100	93,900	224,000	145,000	102,000
29..	17	160,000	133,000	91,600	152,000	31,200	55,600	130,000	217,000	77,900
May 6..	18	68,400	134,000	90,800	142,000	47,000	37,000	76,200	156,000	127,000
13..	19	57,600	179,000	86,500	96,700	42,000	31,200	71,500	90,100	211,000
20..	20	57,600	177,000	128,000	63,400	37,300	27,000	47,600	75,200	193,000
27..	21	39,800	73,500	122,000	56,900	29,500	39,200	33,100	79,200	181,000
June 3..	22	37,800	64,900	139,000	132,000	25,900	31,800	36,200	81,600	146,000
10..	23	32,300	97,200	108,000	114,000	16,800	30,000	99,000	158,000	79,300
17..	24	25,100	65,000	76,400	63,200	15,100	24,600	61,900	145,000	51,500
24..	25	25,500	58,200	69,600	65,000	13,100	21,800	74,900	111,000	43,600
July 1..	26	26,800	38,100	41,100	36,100	13,900	38,200	58,900	105,000	54,600
8..	27	22,600	30,600	45,600	32,600	18,000	22,800	32,900	153,000	64,200
15.	28	20,700	43,300	37,200	34,600	15,600	18,400	25,300	90,500	51,100
22..	29	28,300	41,500	35,500	44,900	15,200	17,600	29,100	54,900	43,500
29..	30	58,000	39,200	43,900	31,600	14,900	14,000	28,900	47,700	36,700
Aug. 5..	31	33,700	31,600	38,700	22,500	11,000	21,000	24,400	45,200	31,700
12..	32	35,000	27,100	40,800	25,900	10,300	39,200	31,500	35,000	35,600
19..	33	55,000	19,600	77,000	24,700	9,450	28,000	31,000	32,000	24,400
26..	34	59,100	19,300	46,300	18,000	7,080	50,600	33,200	78,500	20,000
Sept. 2..	35	37,500	25,300	32,900	17,200	6,570	58,900	23,400	49,300	17,300
9..	36	25,300	21,200	30,500	19,300	5,760	36,300	18,300	95,000	16,900
16..	37	22,600	16,800	25,900	18,100	4,550	25,100	17,300	95,100	22,900
23..	38	19,300	16,400	19,100	16,600	5,790	20,100	15,300	42,800	47,200
30..	39	20,500	14,500	16,200	18,000	7,510	15,800	13,200	33,000	34,300
Oct. 7..	40	23,200	11,800	17,800	33,600	6,480	14,300	12,800	30,400	51,500
14..	41	21,200	11,400	14,300	24,400	7,540	16,100	15,500	27,600	51,600
21..	42	16,100	12,400	17,600	21,400	21,600	14,100	14,600	37,700	23,500
28..	43	14,500	12,800	14,500	14,200	45,500	14,400	14,700	53,000	30,100
Nov. 4..	44	16,200	11,600	13,100	13,500	53,000	16,200	11,800	46,100	58,000
11..	45	21,200	11,500	16,000	13,600	38,100	18,000	13,300	29,200	108,000
18..	46	25,600	11,700	16,000	9,760	125,000	36,300	18,800	24,300	121,000
25..	47	60,800	11,500	15,600	9,870	67,300	74,900	51,800	78,700	225,000
Dec. 2..	48	49,300	10,900	23,200	18,100	41,300	67,000	30,800	66,900	135,000
9..	49	91,500	22,100	45,600	27,900	39,200	67,400	28,200	43,800	66,800
16..	50	57,000	40,100	86,100	86,500	32,800	99,100	58,500	34,400	55,400
23..	51	34,200	145,000	97,400	43,900	50,600	169,000	116,000	33,200	63,000
31..	* 52	63,000	107,000	114,000	27,700	46,800	279,000	69,400	34,400	71,400
Mean for year cfs.......		62,800	82,700	81,700	67,800	39,600	53,700	82,300	76,500	97,300

* 8-day averages.

The average reservoir area is 200,000 acres. The mean annual evaporation is 45 in. The loss due to seepage is estimated at 10 cfs per day.

Required:

a) A flow-duration curve corrected for losses.
b) A mass curve of corrected yield for the three driest years in the period.
c) The regulated flow from the reservoir in cubic feet per second, assuming that the maximum allowable drawdown is 4 million acre-ft at the end of the driest period (the end of the 41st week of 1925).
d) A power-duration curve for the year 1925, assuming an average efficiency of 88 per cent, neglecting the effect of drawdown in the reservoir.

3–2. Procure data, similar to that given in table for Problem 3–1, from published reports of federal or state agencies. Make reasonable assumptions as to seepage and evaporation losses and solve this problem as indicated in Problem 3–1.

3–3. Procure information from your local electric power source which information will be required to plot a load curve for a critical weekly period. Compute the load factor from the plotted curve.

3–4. A run-of-river plant is located on River A and a reservoir plant is located nearby on River B, a tributary of River A entering above the plant on River A. On a certain day the load is as shown in Fig. 3–7, page 56, the flow in River A is 500 cfs and the head on the run-of-river plant is 60 ft. An average of 350 cfs may be released from the reservoir on River B through its plant, which develops a net head of 200 ft. There is to be no waste of water. Assume efficiency at 88 per cent.

Required:

a) The pondage in acre-feet required at the plant on River A.
b) The type and number of turbines to be installed at each plant.

3–5. *Note:* This problem is not a practical one but serves as a review of hydrology and flood routing.

The Northern Power Co. is considering the power generation due to the runoff of the basin in the Silver Creek Power Development. The storm of January 18 to February 1, 1935 is investigated.

The installation consists of a number of turbines, each of which will develop the full power from 2000 cfs at 200-ft rated head. Neglect the loss in efficiency due to the turbines carrying partial load. Use 88 per cent efficiency at all heads.

When the reservoir is at elevation 1014 the storage is greatest. The increase in elevation above spillway crest (elevation 1000 M.S.L.) is directly proportional to the increase in storage.

It is required to know:

a) What was the depth of excess rainfall?
b) A distribution hydrograph and a unit graph.
c) How much power would be produced out of the excess rainfall?

d) Assuming that the base flow of 2000 cfs also produced power during the period of disposal of the excess rainfall, how much power was produced by the base flow?
e) What is the total power in kilowatt-hours?
f) How many turbines are required for the peak load?
g) The details of turbine installations and a sketch of the plan and elevation of the powerhouse.

Data—Drainage area: 1650 sq mi Base flow: 2000 cfs

Average Daily Reservoir Inflow

Jan. 18	2,000 cfs
19	4,540
20	16,500
21	17,500
22	18,500
23	10,300
24	5,690
25	4,280
26	3,530
27	3,170
28	2,950
29	2,620
30	2,510
31	2,200
Feb. 1	2,000

Average Storage–Outflow Relation

Storage (sfd)	Outflow * (cfs)
100,000 (El. 1000) ..	2,000
100,840	4,000
102,750	5,000
105,000	5,800
108,600	7,000
112,600	8,000
115,000	8,500
117,000	9,000
120,000	9,600
122,000	10,000
125,000	10,600
130,000	11,450

* Base flow is included.

CHAPTER 4

4–1. Estimate the average values of WR^2 for generators of the following plants, using the formula given in Art. 4–6 and values of kva and rpm listed in Table 5–1: Kentucky Project, Pickwick Landing Project, Wheeler Project, Guntersville Project, Chickamauga Project, Norris Project, Cherokee Project, and Apalachia Project.

4–2. Select a typical existing hydro power plant and study its technical report. Supposing that you were the design engineer, make a list of information to satisfy the turbine manufacturer for making a proposal to furnish hydraulic machinery.

4–3. The rated head at Wheeler Project is 48 ft and the normal speed is 85.7 rpm. What will be the change in turbine speed due to a fluctuation of 2 ft in head?

4–4. The Chickamauga plant of the Tennessee Valley Authority is operated under a rated head of 36 ft and is at elevation of 628.0 (minimum tailwater elevation above M.S.L.). Assuming an average water temperature of 50°F, estimate the theoretical elevation of center line of the distributor. (N_s = 161)

4–5. Calculate the flywheel constant for one turbine at Grand Coulee Dam which develops 150,000 hp at 325-ft net head at a speed of 120 rpm. The WR^2 of the turbine is 4,270,000 and of the generator is 174,200,000.

4–6. Calculate the relative speed change for a 50 per cent drop in power from full load under the following conditions: $P = 55{,}000$; $WR^2 = 59{,}400{,}000$; $N = 100$; net head $= 136$; governor time 4 seconds.

4–7. A plant is located at an elevation of 300 ft M.S.L.; the average water temperature is 60°F. The turbines develop 50,000 hp at a rated head of 135 ft and run at 112.5 rpm. Calculate the minimum allowable distance from minimum tailwater to center line of distributor.

4–8. What difference in minimum distance from minimum tailwater to center line of distributor would result if the plant in Problem 4–7 were located at 5000 ft M.S.L. and the average water temperature 50°F?

4–9. A fixed-blade propeller turbine is operating at 50 per cent full load and producing 10,000 hp, at a 40-ft head. With the same amount of water and same head, how much power would an adjustable-blade propeller turbine be producing if it were also operating at 50 per cent full load? (See Fig. 4–2.)

CHAPTER 5

5–1. Determine by the aid of the diagram of Fig. 5–14 the approximate generator diameters of plants listed in Table 5–1 and check the results with the installed diameters.

5–2. Estimate the approximate superstructure space requirements for the following hydroelectric plants built by the Tennessee Valley Authority: The Wheeler Project, the Pickwick Landing Project, the Cherokee Project, and the Fort Loudoun Project. Check the results with the actual dimensions of these plants described respectively in the Technical Reports Nos. 2, 3, 7, and 11 of the Tennessee Valley Authority.

5–3. A hydro plant operating under a head of 35 ft has two units of a total capacity of 50,000 hp. The efficiency of the plant is 80 per cent. By the method of constant angular velocity, make a preliminary design of the turbine casing including the following items:

a) Type and material of the turbine casing.
b) Design velocity of flow in the casing.
c) A sketch drawn in scale showing the approximate dimensions of the plan and profile of the casing.

5–4. Calculate D_1 and D_3 for each of the Francis turbine runners shown in Figs. 2–1A, 2–1B, and 2–1C from the data given in the captions. Check the ratio of D_3/D_1 for each of the turbines with the ratio given by Eq. (2–5).

5–5. Calculate the discharge diameter of the Kaplan runner shown in Fig. 2–2A from the data given in the caption. What is the model ratio?

5–6. Estimate the diameter of the impulse runners shown in Fig. 2–3. Remember that each runner is rated at 20,000 hp. The actual diameter is 115 in.

5–7. From data given in Table 5–1 check the given runner discharge diameters from the information given in the text for such plants as may be assigned by the instructor.

5–8. The following table gives actual important dimensions for certain hydroelectric plants. Check these dimensions from information given in the text and in Table 5–1. The numbers in Column 1 of the table refer to the numbers designated to the plants in Column 1 of Table 5–1.

DIMENSIONS OF EXISTING PLANTS

Plant No.	Scroll Case					Draft Tube			
	C	F	G	I	J	L	H	P	CW
2	19.0	29.3	26.5	22.9	17.6	59.1	37.0	15.8	42.0
3	12.0	22.5	21.1	19.1	16.1	50.0	35.0	12.5	39.5
8	15.0	28.6	25.8	22.9	18.6	54.5	37.9	15.8	53.0
10	12.7	22.0	19.2	17.0	14.2	56.5	30.7	14.0	48.3
13	11.0	21.0	18.9	16.8	13.6	59.0	28.3	11.6	45.0
14	8.5	14.3	12.9	11.4	9.3	25.3	15.4	7.9	24.0
24	18.0	27.3	24.7	21.5	16.9	55.5	33.0	14.0	41.4
25	20.0	38.0	35.0	31.5	26.5	59.0	39.0	15.0	44.8
30	14.7	23.8	21.8	19.2	15.7	46.4	33.0	13.0	44.0
34	12.5	19.7	17.9	15.3	12.1	40.0	24.0	10.0	25.5
44	19.8	29.9	27.1	23.7	19.2	63.0	42.0	16.0	50.6
46	69.0*	39.8	31.9	29.3	....	85.0	64.6	28.0	56.0
48	66.0*	36.0	29.7	30.0	....	85.0	60.0	25.5	54.0
50	57.0*	30.5	25.2	25.5	....	73.0	50.0	21.5	45.0
51	62.5*	34.8	36.0	23.8	....	85.0	57.0	24.0	51.0
58	74.0*	43.3	35.3	24.1	....	105.0	59.5	24.0	73.0

* Intake width including two piers each 6.5 ft in width.

5–9. Compare the dimensions of the Wheeler scroll case (Table 5–2, page 85) with information given in text. The runner discharge diameter is 22 ft.

CHAPTER 6

6–1. In making a preliminary study on speed regulation of a hydroelectric unit, the following data are available:

Head: 48 ft
Turbine capacity: 45,000 hp
Type of runner: encased propeller
Normal speed of the unit: 85.7 rpm
Flywheel effect: 4,500,000 lb-ft sq for turbine
61,260,000 lb-ft sq for generator

Permissible speed increase on sudden dropping of the full load: 30 per cent

Required:

a) Estimate the governor servomotor capacity.
b) Compute the required time of closure for the governor.
c) As a general rule, the governor time should not be less than 3 seconds. Is this rule satisfied in the present case?
d) If the maximum speed increase is limited to 20 per cent, is the above rule satisfied? If not, then what would you do? Give a numerical answer.
e) A servomotor consisting of two cylinders, each having a bore diameter of 1 ft; diameter of piston rod 1 in., and stroke 5.5 ft, is in stock. Can this servomotor be used for the governor? The standard normal maximum oil pressure in modern governors of large size is 300 psi.
f) If the above servomotor could be used, then compute the capacity in gallons per minute of the oil pump required to drive the piston.
g) Draw a sketch of a turbine governor and describe its operation.

6–2. In making a preliminary study on speed regulation of a hydroelectric unit, the following data are available:

Head: 400 ft
Turbine capacity: 105,000 hp
Type of runner: Francis
Outlet diameter of runner: 10.5 ft
Normal speed of the unit: 180 rpm
Flywheel effect: 1,100,000 lb at 1 ft radius for turbine
68,900,000 lb at 1 ft radius for generator

Permissible speed increase on sudden dropping of the full load: 30 per cent

Required:

a) Estimate the governor capacity. (Use outside type of gate-operating mechanism.)
b) Determine the flywheel constant or regulating constant for the unit to check the stability of regulation.
c) Compute the required time of closure for the governor.
d) As a general rule, the governor time should not be less than 3 seconds. Is this rule satisfied in the present case?
e) A servomotor consisting of two cylinders, each having a bore diameter of 13 in., diameter of piston rod 2 in., and stroke 3.0 ft, is in stock. Can this servomotor be used for the governor? The standard maximum oil pressure in a modern governor of large size is 300 psi.
f) If the above servomotor could be used, then compute the capacity in gallons per minute of the oil pump required to drive the piston.

6–3. Compute the pressure rise in the penstock of the hydroelectric unit in the previous problem in order to conform with the computed governor time. Does this pressure rise exceed the permissible value? What is the corresponding increase in the normal speed of the turbine? The penstock is 412 ft in length and 13.5 ft in diameter.

6–4. Using the Kirschmer formula, estimate the loss of head, in terms of the velocity head, through the racks of the Shasta Dam trashrack structure as shown in Fig. 6–27.

6–5. An air inlet is provided at the summit of the penstock described in Problem 6–3. The penstock has a wall thickness of ¾ in. and carries a maximum discharge of 800 cfs. The air inlet should be designed for a flow of air equal to the maximum flow of water in the conduit. The safe difference in pressure P between inside and outside of pipe depends on the strength of steel pipe which may be calculated by the following formula:

$$P = \frac{50{,}200{,}000}{s}(t/d)^3$$

in which t = thickness of steel pipe, in inches
d = diameter of steel pipe, in inches
s = a factor of safety against collapse of pipe

Assuming a factor of 10, compute the area required for the air inlet.

6–6. A penstock to serve a single unit which develops 11,200 hp under a net head of 215 ft (at the entrance to the turbine) is constructed as follows: A horizontal welded steel pipe 12 ft in length with inside diameter of 96 in. is connected with a similar sloping pipe (19°55′38″) 394 ft in length; this pipe is connected with a similar sloping pipe (45°0′) 94 ft 9 in. in length. The final pipe is 32 ft in length with inside diameter of 72 in. and is laid horizontally. The elevation of the center line of the turbine is 3031 M.S.L. Elevation at forebay is 3249 M.S.L. Design (a) the penstock, assuming an allowable stress of 14,000 psi, minimum plate thickness ⅜ in., governor time 5 sec., R/D greater than 2; (b) the anchor at the bend between the 19°55′38″ pipe and the 45° pipe. Expansion joints are located 200 ft above the bend and 23 ft below the bend. Three piers lie between the bend and the upper expansion joint. There are no piers between the bend and the lower expansion joint. Coefficient of friction of pipe on piers is 0.5. Make reasonable assumptions for any other data that may be necessary.

CHAPTER 8

8–1. The Northwestern Power Co. is considering the development of a hydro site on Grand River. The proposed plant would be operated as a part of an interconnected system which is also supplied with steam power. Field investigations and studies made available the following data:

Elevation: 2000 M.S.L. Average Temperature 55°F

Flow Data:

% time:	1	5	10	20	30	40	50	60	70	80	90	97	100
Flow cfs/mi²:	8.0	6.0	5.0	3.5	2.2	1.8	1.4	1.0	0.8	0.7	0.6	0.5	0.3

Drainage area: 11,000 sq mi   Average net effective head: 65 ft
Pondage: The site lends itself to development of adequate facilities.

Load Data:

Average daily load in per cent of peak

Hours:	1	2	3	4	5	6	7	8	9	10	11	12 o'clock
A.M.:	24	20	8	5	5	20	56	72	70	66	56	60%
P.M.:	60	62	64	70	80	100	80	60	52	44	42	24%

Power factor: Assume as 1.0 in all cases
Peak load: 265,000 kw
Generator: 60 cycles, 3 phase, alternator
Transmission lines: 25 miles long operated at 33,000 v; line loss 4.5 per cent
Hydro plant over-all efficiency: 88 per cent

Cost Data:

Hydro
Installation: See Fig. 7–1. Fixed charges: 9.5 per cent
Operation cost: See Fig. 7–2.
Transmission: $2.00 per kw of installed capacity per year

Steam installation: See Fig. 7–3.
Fixed charges: 12 per cent
Fuel: 12,000 Btu per lb; cost $4.00 per ton
Plant performance: 11,000 Btu per kwh
Operation and supplies: See Fig. 7–4.

Required: A report containing: all calculations; all necessary curves; sketches showing general plan of development; plan and cross section of powerhouse showing controlling dimensions; notes on your selection of equipment; a brief statement of your assumptions and your engineering opinion of the feasibility and general aspects of the project. Your opinion as to feasibility should be supported by a brief economic analysis. The entire report is to be neatly bound in a folder. Pencil may be used for the preparation of sketches and curves and for any other part you may desire.

8–2. The Eastern Power Co. is considering the development of a hydro site on Duaine River. The proposed plant would be operated as a part of an interconnected system which is also supplied with steam power. Field investigations and studies made available the following data:

Elevation: 1000 M.S.L.   Average Temperature 60°F

Flow Data:

% time:	1	5	10	20	30	40	50	60	70	80	90	97	100
Flow cfs/mi²:	10.0	7.0	6.0	4.2	3.1	2.3	1.9	1.3	1.0	0.9	0.7	0.6	0.5

Drainage area: 7500 sq mi   Average net effective head: 40 ft
Pondage: The site lends itself to development of adequate facilities.

Load Data: The data given below represent the load demand on the power system during a week in December when the load is at a maximum, and the flow for the run-of-river plant is at a minimum.

TABLE 3–1. LOAD IN 1000 KW.

Time:	Sun.	Mon.	Tues.	Wed.	Thurs.	Fri.	Sat.
			A.M.				
1	42	38	60	56	58	54	57
2	38	38	56	60	59	52	56
3	36	43	57	60	58	48	54
4	39	42	54	58	57	50	52
5	33	42	53	54	55	50	53
6	34	40	52	52	52	50	54
7	34	72	68	70	69	70	68
8	38	89	87	91	92	90	83
9	36	92	96	94	94	93	84
10	38	96	96	100	96	96	86
11	36	98	100	100	98	98	86
12	38	98	94	96	94	97	76
			P.M.				
1	38	80	80	83	79	82	67
2	36	92	98	92	96	94	70
3	38	94	100	96	96	94	67
4	37	96	98	96	96	94	68
5	36	86	88	91	90	86	58
6	38	68	69	68	67	70	50
7	39	62	64	64	61	68	52
8	54	70	74	74	71	70	60
9	56	74	74	72	70	72	65
10	52	70	69	66	68	68	60
11	50	64	62	67	60	64	54
12	44	61	64	65	54	62	50

Required:

a) Plot a flow-duration curve.
b) Plot a power-duration curve (scales 1 in. = 20,000 kw and 1 in. = 20 hr).
c) Plot a load-demand duration curve.
d) Determine an auxiliary scale which indicates the per cent of area (for each 5 per cent) which lies below the load demand duration curve and above a horizontal line.
e) Determine the load factor for the system.

8–3. Basic Data: All data given in Problem 8–2 with the following additions:

Steam
Unit cost installation: See Fig. 7–3.
Fixed charges: 13½ per cent.
Fuel: 12,000 Btu per lb of coal; cost $4.80 per ton.
Performance: 15,000 Btu per kwh.
Operation and maintenance: See Fig. 7–4.

Hydro

Unit cost installation: See Fig. 7–1.
Fixed charges: 9 per cent.
Annual operation and maintenance: See Fig. 7–2.

Required:

a) Curves showing total production cost in mills per kwh for hydro and steam for load factors from 10 per cent to 100 per cent.
b) Load factors for hydro, steam, and system.
c) A statement regarding the economic feasibility of the hydro project with reasons why it should or should not be developed.
d) Neatly prepared sketches showing plan and cross section of the hydro plant with all dimensions clearly indicated. Assume topographic conditions and plant arrangement.
e) Notes on your selection of hydro equipment.
f) A résumé of your studies on the complete project. The entire report to be bound in a cover. Inked drawings are *not* necessary.

8–4. *Design Data:*

Maximum head	180 ft
Minimum head	75 ft
Normal operating head	140 ft
Design maximum flood flow	300,000 cfs
Width of each spillway gate	40 ft
Spillway discharge coefficient	3.86
Top of earth dam, elevation	1120
Deck of non-overflow section, elevation	1102
Crest elevation of spillway	1079
Maximum H. W. elevation	1105
Normal pool, elevation	1103
Maximum tailwater, elevation	980
Minimum tailwater, elevation	923
Discharge channel bottom elevation	920
Generator room floor, elevation	956
Foundations: Excellent rock	

Required:

a) Flow-duration curve. Use Table 3–1.
b) Selections of turbines and generators, giving number, capacity, and speed.
c) Dimension of penstock, scroll case, and draft tube.
d) Cross sections: through powerhouse and non-overflow section of earth dam; and spillway, showing suggested gate controls, trashracks, energy dissipating device, sluiceways, etc.
e) Capacity of governors in ft lb.
f) Other pertinent design information.
g) Plans and front elevations of powerhouse, dam, and spillway.

Note: Show all pertinent dimensions on all plans and cross sections.

8–5. Prepare a report on the Tennessee Valley Authority's most recent program on hydro-steam association, or such parts of the program to be assigned by the instructor.

8–6. Prepare an abstract of "Economic Aspects of Energy Generation," *Trans. A.S.C.E.*, Vol. 104 (1939), and "Cost of Energy Generation," *Trans. A.S.C.E.*, Vol. 104 (1939). Make an approximate estimate of the effect which the differences in prices of labor and materials between 1939 and the present day would have on the values developed in these papers.

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(References to illustrations are indicated by italics)